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THE

STEAM ENGINE CATECHISM:

A SERIES OF

DIRECT PRACTICAL ANSWERS TO DIRECT PRACTICAL QUESTIONS.

MAINLY INTENDED FOR

YOUNG ENGINEERS

AND FOR

EXAMINATION QUESTIONS.

BY

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"Engineer's Hourly Log Book," "Engine
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TO

THE NATIONAL ASSOCIATION

OF

STATIONARY ENGINEERS,

IN APPRECIATION OF ITS EFFORTS TO ADVANCE THE PRACTICE OF ENGINEERING, AND FURTHER THE INTERESTS OF ENGINEERS AND THEIR EMPLOYERS THROUGHOUT OUR ENTIRE COUNTRY, THIS LITTLE VOLUME IS DEDICATED, IN THE HOPE THAT IT MAY PROVE USEFUL.

ROBERT GRIMSHAW,

(PRESIDENT AND DELEGATE, JAMES WATT ASSOCIATION, NO. 7, OF N. Y.)

PREFACE TO FIRST EDITION.

THIS book is intended to give correct, straightforward answers not only to such direct questions as have been asked by both theoretical and practical men, but to questions chosen with the view of saving the reader a long search after some point, and enabling him to get directly at the information desired. Formulas and "mathematical gymnastics" are avoided. The Index will be found to save time.

While the book is not written for professional men, I believe it to be technically correct, and up to date.

I should be glad to receive for reply queries concerning any matter relating to the steam engine; and where possible will not only answer them by mail but embody questions and answers in future editions or volumes, should they be called for.

ROBERT GRIMSHAW.

July 10, 1885.

PREFACE

TO THE

FOURTH EDITION.

This Fourth Edition is called for within a year of the appearance of the first, more by reason of the real scarcity of books of this class, upon this subject, and of the great number of readers desirous of increasing their knowledge in this line, than from any special intrinsic merit in the work itself.

I have thought it best to increase its size by about one fourth; the additional questions being, like all those which are found in the Supplement or Second Volume, either those which have been asked me by mail, and at the meetings of associations of practical engineers, or those which have been suggested by such queries.

All of the Preface to the First Edition is applicable to the present.

New York, July, 1886.

STEAM ENGINE CATECHISM.

DEFINITIONS AND PRINCIPLES.

Q. What is steam?

A. A thin, condensible vapor, or fluid, elastic under compression, and which is one of the three states (ice, water, steam) in which water exists.

Q. What is one of the most marked characteristics of steam?

A. One of the most marked characteristics of steam is its tendency to expand, decreasing its pressure about in the same ratio as it increases its volume.

Q. What is the temperature of steam?

A. The temperature or sensible heat of steam varies with the conditions. Steam formed from water under the ordinary atmospheric pressure (14.7 lbs. per square inch at sea level) has a temperature of 212° F., = 100° C. Confined so that expansion is

impossible, it is generated at a temperature corresponding to the pressure or tension, and may be maintained at that temperature.

We give a table showing the temperature corresponding to various tensions; the tensions being expressed in pounds per square inch, but above vacuum, not above atmosphere.

PRESSURE AND TEMPERATURE OF SATURATED STEAM.
(Regnault.)

Absolute pressure per sq. inch.	Temp. deg. Fahr.	Absolute pressure per sq. inch.	Temp. deg. Fahr.	Absolute pressure per sq. inch.	Temp. deg. Fahr.
1	102.1	15	213.1	30	250.4
2	126.3	16	216.3	31	252.2
3	141.6	17	219.6	32	254.1
4	153.1	18	222.4	33	255.9
5	162.3	19	225.3	34	257.6
6	170.2	20	228.	35	259.3
7	176.9	21	230.6	36	260.9
8	182.9	22	233.1	37	262.6
9	188.3	23	235.5	38	264.2
10	193.3	24	237.8	39	265.8
11	197.8	25	240.1	40	267.3
12	202.	26	242.3	41	268.7
13	205.9	27	244.4	42	270.2
14	209.6	28	246.4	43	271.6
14.7	212.	29	248.4	44	273.

Pressure and Temperature of Saturated Steam.—Regnault.—Continued.

Absolute pressure per sq. inch.	Temp. deg. Fahr.	Absolute pressure per sq. inch.	Temp. deg. Fahr.	Absolute pressure per sq. inch.	Temp. deg. Fahr.
45	274.4	72	304.8	99	327.1
46	275.8	73	305.7	100	327.9
47	277.1	74	306.6	101	328.5
48	278.4	75	307.5	102	329.1
49	279.7	76	308.4	103	329.9
50	281.	77	309.3	104	330.6
51	282.3	78	310.2	105	331.3
52	283.5	79	311.1	106	331.9
53	284.7	80	312.	107	332.6
54	285.9	81	312.8	108	333.3
55	287.1	82	313.6	109	334.
56	288.2	83	314.5	110	334.6
57	289.3	84	315.3	111	335.3
58	290.4	85	316.1	112	336.
59	291.6	86	316.9	113	336.7
60	292.7	87	317.8	114	337.4
61	293.8	88	318.6	115	338.
62	294.8	89	319.4	116	338.6
63	295.9	90	320.2	117	339.3
64	296.9	91	321.	118	339.9
65	298.	92	321.7	119	340.5
66	299.	93	322.5	120	341.1
67	300.	94	323.3	121	341.8
68	300.9	95	324.1	122	342.4
69	301.9	96	324.8	123	343.
70	302.9	97	325.6	124	343.6
71	303.9	98	326.3	125	344.2

Pressure and Temperature of Saturated Steam.—Regnault.—Continued.

Absolute pressure per sq. inch.	Temp. deg. Fahr.	Absolute pressure per sq. inch.	Temp. deg. Fahr.	Absolute pressure per sq. inch.	Temp. deg. Fahr.
126	344.8	145	355.6	240	397.5
127	345.4	146	356.1	245	401.1
128	346.	147	356.7	260	404.5
129	346.6	148	357.2	270	407.9
130	347.2	149	357.8	280	411.2
131	347.8	150	358.3	290	414.4
132	348.3	155	361.	300	417.5
133	348.9	160	363.4	350	430.1
134	349.5	165	366.	400	444.9
135	350.1	170	368.2	450	456.7
136	350.6	175	370.8	500	467.5
137	351.2	180	372.9	550	477.5
138	351.8	185	375.3	600	487.
139	352.4	190	377.5	650	495.6
140	352.9	195	379.7	700	504.1
141	353.5	200	381.7	800	519.5
142	354.	210	386.	900	533.6
143	354.5	220	389.9	1000	546.5
144	355.	230	393.8		

Q. Is steam estimated by weight or by volume?

A. Steam should always be estimated by weight.

Q. What is saturated steam?

A. Saturated steam is that which is generated "at the maximum density and pressure corresponding to its temperature. It is incapable of vaporizing more water into the same space unless the temperature be raised. Saturation is therefore the normal condition of steam generated in contact with a store of water, and the same density and the same pressure are always to be found in conjunction with the same temperature." (D. K. Clark.) It is condensed if the temperature falls, and more water is evaporated if the temperature rises.

Q. Which is the hotter, dry steam or saturated?

A. Dry steam is no hotter than saturated.

Q. How much steam will one cubic inch of water make?

A. That depends on the temperature and pressure. At 212° F = 100° C, one cubic inch of water will make 1641.5 cubic inches at the atmospheric pressure of 14.7 lbs. per square inch.

Q. What is the function of steam in the steam engine?

A. The function of steam in the steam engine is to transform the heat of combustion into mechanical work; a task which, for various reasons, it accomplishes very imperfectly.

Q. What are the causes which prevent the utilization as mechanical work of any considerable portion of the energy of combustion?

A. The causes which limit the utilization of the energy of combustion, as mechanical work, are as follows:

Imperfect combustion.

Imperfect transfer of the heat of combustion to the water and steam in the boiler.

Incomplete utilization of the heat of the steam by the engine.

Friction and back pressure in the engine.

LOSSES OF HEAT, ETC.

Q. How may the various losses and wastes in the boiler and engine be apportioned?

A. We may charge the chimney with 25 per cent., the fireman or "stoker" with 10

per cent., and the loss by exhaust steam 55 per cent.; leaving only about 10 per cent. accounted for by the indicator as being converted into work.

Q. How many horse-power can we get by the consumption of one pound of coal per hour?

A. In practice, the consumption of one pound of coal per hour, under the most favorable conditions, develops one-half indicated horse-power.

One-third horse-power per pound of coal per hour is considered excellent work; and one-sixth is all that is often obtained.

Q. How many pounds of water will one pound of coal evaporate?

A. The weight of water that one pound of coal will evaporate depends on the temperature of the water and the pressure and temperature of the steam made from that water; also on the thoroughness and rapidity of combustion, the kind of boiler, etc.

Each pound of the best coal should evaporate 15 pounds of water from and at 212° F.

It generally does about half that, and seldom over 10 pounds, although 12 is sometimes reported.

The following figures show the evaporation in the boilers of the West-side Water Works, Chicago (Henry Mason, Chief Engineer).

	Lbs. of Water per lbs. Coal.
Laurel Hill, lump	12.09
Lackawanna, hard	10.98
Erie, lump	10.95
Pittsburgh, nut	10.47
Massilon, O., lump	10.28
Indiana Block, lump	9.55
Wilmington, lump	8.59
Streator, nut	8.09
Gaston, W.Va., lump	10.24
Piedmont, W.Va., lump	11.72

The feed water was heated to 212° by exhaust from the feed pumps, and the water weighed into tanks.

All of these tests were made continuously, one after the other, and in regular actual daily duty.

The parties furnishing the different coal

in all cases were present during these trials, either in person or representative. Steam pressure 70 lbs. by the gauge. Rocking grate was used.

What are the sources of loss of heat in a steam engine?

A. The sources of loss of heat in a steam engine are radiation, condensation, and exhausting steam at a high temperature and pressure into the air.

Q. How may radiation from boiler, pipes, and engine cylinders be prevented?

A. Radiation from boiler, pipes and cylinder may be prevented by covering them with a poor conductor of heat; and radiation from the steam cylinder may be lessened by surrounding it with a "steam-jacket," containing either live or exhaust steam. The addition of a steam jacket is expensive, particularly so, in comparison, on small cylinders. Jacketed cylinders should be covered with a non-conducting substance.

The following table gives the relative values of certain non-conducting coverings, as shown by tests made by the author:

	Average Pressure of Steam, lbs.	Average Temperature of Steam Pipe, Fahr.	Average Temperature of Covering.	Average Temperature of Room.	Relative Non-conducting Values of Covering.
Asbestos and Hair Felt, 1 in. thick, 1 in. air space.	$36\frac{1}{4}$	$268\frac{1}{16}$	$115\frac{1}{4}$	90	1.0000
Asbestos and Hair Felt, 1 in. thick, no air space..........	$41\frac{1}{8}$	$264\frac{3}{8}$	$127\frac{1}{8}$	91	0.8952.
Sectional Plaster, 1 in. thick, 1 in. air space.	44	$259\frac{3}{8}$	$173\frac{3}{8}$	85	0.7438
Asbestos Cement, 1 in. thick, no air space..	$47\frac{5}{16}$	270	$158\frac{1}{16}$	$91\frac{7}{16}$	0.8039

Q. What is a steam-jacket?

A. A steam-jacket is a metallic steam tight casing, leaving an annular space between it and the cylinder walls, filled with live or exhaust steam.

(The steam-jacket is thoroughly treated of under "Economy," etc.)

Q. How may internal cylinder condensation be lessened?

A. Internal cylinder condensation may be lessened by superheating the steam, or by jacketing the cylinder; also by avoiding excessive expansion ratios.

(See under heads Superheating, Jacketing, Expansion, Economy.)

Q. What proportion of the heat of the steam is carried away in the exhaust?

A. The proportion of heat carried away in the exhaust varies with the initial and terminal pressures,* and with the piston speed and rate of revolution. The higher the initial pressure, the greater the compression or "cushion," and the higher the piston speed and rotation speed, the less the proportion of loss of heat by the exhaust.

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* See under these heads.

Q. May the heat be recovered from the exhaust steam of an engine?

A. The heat cannot be recovered from the exhaust without interfering with the performance of the engine.

Q. How may loss of heat in the exhaust be lessened?

A. Loss of heat in the exhaust may be lessened by compression or cushion, and by high speed of piston and shaft.

Q. How may some of the heat of the exhaust be utilized?

A. Some of the escaping heat of the exhaust may be utilized by heating the boiler feed water,* or by heating rooms, wort, dye, etc., drying material, and so on.

Q. How much power is absorbed by a good engine in overcoming the friction of its own parts?

A. The engine friction proper varies with the excellence of design and workmanship, and also with the load on the engine.

At the Cincinnati engine trials of 1880 the frictions of the three engines tested

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*See under "Economy."

(non-condensing) were as follows, other elements being given:

	Renolds Corliss.	Harris Corliss.	Wheel-ock.
Boiler Pressure, lbs.........	96.61	96.32	96.22
Pipe Pressure.	92.54	91.48	91.54
Revolutions, per min........	75.330	75.810	76.072
Initial Pressure	89.994	89 522	88.499
Cut off at....	.15956	.13627	.16966
Pressure at cut off...........	84.757	85.910	76.875
Terminal Pressure of.	17.418	17.087	17.460
Back Pressure, mid-stroke..	.945	.415	.957
Maximum Compression.....	84.704	46.098	44.218
Mean Effective Pressure...	29.7805	28.9397	29.3632
Indicated Horse Power.....	187.0171	184.2926	189.9726
Friction of Engine, H. P...	10.2552	9.5609	7.8919
Gross Load................ ..	126.7619	124.7317	181.9907
Extra Friction of Engine due to load.	5.0705	4.9893	5.2796
Net effective H. P.	121.6914	119.7424	126.7111

DEFINITIONS OF PRESSURES.

Q. In what direction is the atmospheric pressure exerted?

A. The pressure of the atmosphere is exerted in all directions, and in equal degree in every direction.

Q. How is atmospheric pressure measured?

A. Atmospheric pressure is usually measured by the mercurial barometer, consisting of a tube closed at the top, and in which

the pressure of the air holds up a column of mercury which varies with that pressure. A column of mercury at 32° F., and one square inch in section, and 29.22 inches or 76 centimeters high, weighs 14.7 lbs. A 62° F., the height of the mercurial column weighing 14.7 lbs. per square inch, is 30 inches. A column of water at 62° F., on square inch in area, and 33.947 feet, or 10.347 meters high, also weighs 14.7 lbs.; and this is the maximum theoretical height to which a pump can draft water.

Q. What is the difference between pressure above atmosphere and pressure above vacuum?

A. The ordinary pressure of the atmosphere is 14.7 lbs. per square inch, at the sea level. At higher levels it is less.

Q. What is "piston displacement?"

A. Piston displacement is the volume "swept through" or "generated" by the piston during a complete single stroke.

Q. How is piston displacement calculated?

A. Piston displacement is calculated by multiplying the area of the piston by its stroke expressed in the same kind of units

—thus, area in square inches by stroke in inches, not by stroke in feet.

In a 20 x 48 engine the piston area is 20 × 20 × .7854 = 314.16 * square inches, and the piston displacement is 314.16 × 4 = 15079.68 cubic inches = 8.72 + cubic feet.

Q. What is initial cylinder pressure?

A. The initial cylinder pressure is the pressure in the cylinder at the beginning of the stroke; it should be higher than that at any other point of the stroke. It may be expressed either above vacuum or above atmosphere; and is understood to be the latter unless stated to the contrary. The pressure is generally lowered toward the end of the stroke, in throttling engines, by wire drawing † and the throttling † action of the governor, in all engines having a cut-off by expansion; † and by leakage, cylinder condensation, etc.

Q. What is back pressure or counter pressure?

A. Back or counter pressure is the retarding pressure on the piston, during a stroke, and tending to lessen the effective

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*This is readily found by consulting the table of areas of circles

†See under these heads.

pressure and the work of the engine. In condensing engines it is reckoned above vacuum; and in non-condensing engines either above vacuum or above atmosphere, although the latter is understood if the former is not specified.

Q. How high should the back pressure be?

A. Back pressure should never exceed 3 lbs. by the gauge, in non-condensing engines, and 7 lbs. above vacuum in condensing engines.

Q. How may back pressure be lessened and the power of an engine increased?

A. Non-condensing engines that are wasteful by reason of back pressure, may at times be made economical by the application of a condenser.

The ideal diagram, Fig. 1, shows the proportions of work done by an engine without a condenser and by the condenser. Steam is cut-off at ½ stroke; there is about 4 lbs. back pressure, or 10.7 lbs. of vacuum; the initial pressure is 34.7 lbs. above vacuum, or 20 lbs. by the gauge; the terminal pressure 17.35 lbs. absolute, or 3.65 lbs. by the gauge. The condenser does

about half as much as is done without it, or about one-third of the whole work.

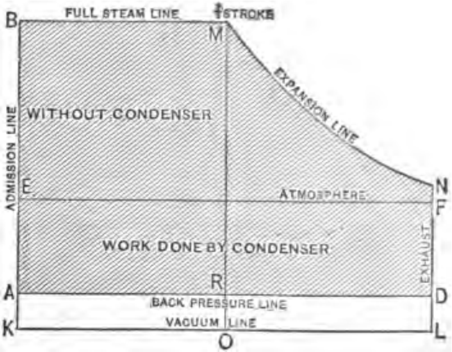

FIG. 1.

Q. Suppose it is not necessary to increase the power of the engine?

A. Then the condenser* will permit lower initial pressure, or slower speed, or earlier cut-off to give the same power.

Q. What is average total pressure?

A. Average total pressure is the average pressure on one side of the piston, during a

* See Condensing.

whole stroke. Owing to throttling, wire-drawing, cylinder condensation, expansion, etc., it is always lower than initial pressure.

Q. How may average total pressure be measured?

A. Average total pressure may be measured by means of an "indicator," which gives a "card" or diagram, the height of which may be measured on the proper scale at several equi-distant points and the average taken.

Q. How may average total pressure be calculated?

A. Average total pressure may be obtained by dividing the stroke in feet into the difference between the clearance in feet and the period of admission plus clearance, times one, plus the hyperbolic logarithm * of the actual expansion rate; and multiplying the quotent by the total initial pressure in pounds per square inch.

As a formula:

$$p = \frac{P\ [l'\ (1 + \text{hyp. log. } R')-c]}{L}$$

For instance: we have initial cylinder pressure 80 lbs., by gauge (that is 94.7 lbs.

* See table of hyperbolic logarithms.

absolute); stroke, one foot; cut off at ¼; clearance, two per cent.; we have the actual expansion rate

$$R' = \frac{1 + .02}{.25 + .02} = 3.78$$

the hyperbolic logarithm of which is 1.3297; and the equation becomes

$$p = \frac{94.7\,[.27\,(1 + 1.2397) - .02]}{1} = 57.67 \text{ lbs.}$$

Q. How may average total pressure be approximated?

A. Average total pressure may be approximated by calculating the pressure at several equidistant points of the stroke, and averaging them.

The theoretical pressure at any point of the stroke, where there is expansion, is practically inversely as the volume. Thus, if the cut-off is at ¼ stroke, the terminal pressure will be ¼ the pressure at cut-off, because the steam fills four times the volume; at ½ stroke, the pressure is ½, because the volume is doubled; at ¾ stroke it is ⅓, the volume being three times as great, etc.

Fig. 2 shows by an ideal diagram, the pressure above vacuum at beginning and at end of stroke, and at ½ and ¾ stroke.

Q. What is mean effective pressure?

A. The mean effective pressure on a piston is the difference between the average

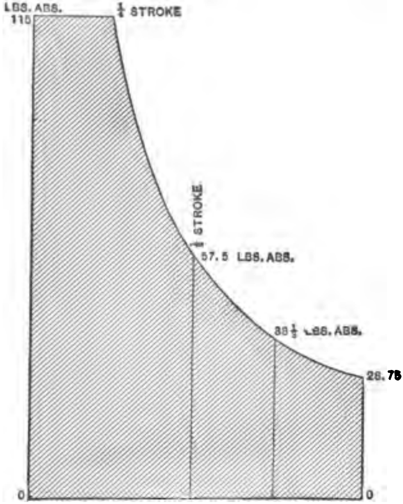

FIG. 2.

total pressure and the average back pressure; both of these latter being reckoned from the same point.

Q. What is final or terminal pressure?

A. Final or terminal pressure is some-

times considered as the actual pressure at the end of the stroke; but as in some cases exhaust takes place before the end of the stroke, the term might be applied to that what would have existed at the end of the stroke had not "release" taken place before that point.

Q. What is absolute pressure?

A. Absolute pressure is pressure reckoned from the zero or absolute vacuum. Initial cylinder pressure, mean total pressure, terminal pressure, and back pressure, all may be either "absolute" (that is above vacuum or zero) or "by the gauge" (which is the same thing as above atmosphere). Mean effective pressure is neither "absolute" nor "by the gauge." It is simply the difference between mean total and mean back pressures.

Q. What is a high pressure engine?

A. The term "high-pressure engine" is a misnomer as applied to present practice. It once meant a non-condensing engine, which required high pressure to be at all economical.

Q. What is a low-pressure engine?

A. A "low-pressure engine" was, in old days, one which used a condenser to reduce

the absolute back-pressure, hence permitting, and in fact almost requiring, the use of lower pressure steam than where the engine was run non-condensing.

CLEARANCE.

Q. What is clearance?

A. "Clearance" is the waste volume between the valve and the piston-head, at the beginning of the stroke.

It is not simply the space between the piston-head and the cylinder-head, but runs up the passage to the valve.

Q. What is the influence of clearance?

A. The influence of clearance is to lessen the actual expansion rate.* The shorter the stroke the greater the influence of a given clearance volume (the piston area being the same.)

It produces a higher terminal pressure for a given cut-off, or a less mean effective pressure for a given terminal.

It also causes loss of steam from the expansion and escape of steam in the clearance volume—when the exhaust takes place at a higher pressure than exists during return stroke.

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* See this head.

Clearance is specially wasteful in condensing engines, as in these there can be no compression.

It may affect the governing, as where the cut-off is early, the steam in the clearance volume being proportionately large to the total volume admitted, tends to speed the engine under light loads.

Figure 3 shows the ill effects of too

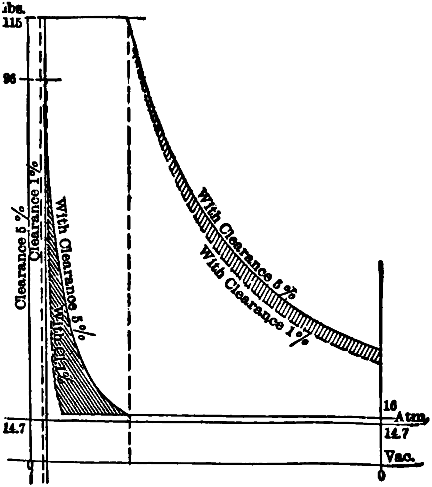

FIG. 3.—EFFECTS OF CLEARANCE.

much clearance, even where there is plenty of cushion. There are two ideal indicator cards † here, one enclosed by a full line, the other partly enclosed by dotted lines. In each there is cushion to initial pressure, but the difference in the cards with 1 per cent. and that with 5 per cent. clearance is very marked.

Q. How may the evil effects of excessive clearance be lessened, and why?

A. The evil effects of excessive clearance may be lessened by closing the exhaust valve before the end of the stroke, thus compressing the steam to as near as possible the temperature and pressure in the steam-chest.

Q. How much clearance is there generally in a steam-engine?

A. Slide-valve engines have more clearance than those with piston, poppet or "plug" valves. The shorter the stroke the greater the *percentage* of clearance. In some short stroke slide-valve engines it will run as high as twelve per cent., and in some long stroke engines with piston, poppet or plug valves, as low as two per cent.

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* See under head of "The Indicator."

Q. Is clearance necessary?

A. It is impossible to construct an engine without some space between the valve and the cylinder. If these passages be given very small sectional area, they wire draw the steam; if too large, they cause waste of efficiency.

Cylinder clearance is left to prevent the piston-head striking the cylinder-head in case of lost motion at crank-pin or cross-head; also in case there should be water in the cylinder. The higher the speed the more cylinder clearance necessary.

The clearances of the three 18 in. x 48 in. engines tested at the Cincinnati engine tests of 1880 were as follows:

Reynolds-Corliss, .0265
Harris " .0193

The Wheelock engine has two clearances, one for compression, .0142 of the piston displacement, and one for expansion, .0235, of the same volume.

THE SLIDE VALVE.

Q. What is a slide valve?

A. A slide valve is a device employed to effect, regulate the admission and discharge of steam from the cylinder or an engine,

and most often consisting of a plate sliding upon a ported seat.

Q. What are the usual shape and style of a slide valve?

A. A slide valve is usually a flat plate having in the centre an arch with closed ends, between two "legs" or "lips" or "faces" having parallel edges. It is generally slid in a direction exactly at right angles to these edges. This style is often called a "D" valve.

Q. What name is given to these edges?

A. The outer ones or ends of the valve are known as the steam edges, and the inner ones, which bound the arch, are the exhaust edges.

Thus referring to Fig. 4, page 36, representing the simplest type of D valve, and also in all the other figures of the same set, we have the following terms :—

AB.—Back leg.

BC.—Arch.

CD.—Front leg.

EF.—Back port.

FG.—Back bridge.

GH.—Exhaust port.

HK.—Front bridge.

KL.—Front port.

(The terms "front" and "back" as used in this case are merely relative. The front part of a horizontal stationary engine is the one nearest the crank; the front port of a locomotive engine is the one farthest from the crank, whether the engine be running ahead or backwards. Vertical engines have neither "front" nor "back" ports; and some inclined engines may have the ports named almost any way).

Q. What is the function or duty of the D valve?

A. The D valve has for its duty, function, or office putting each end of the engine cylinder in communication alternately with live steam in the steam chest, and also with either the open air as a condenser, according as the engine is a non-condensing or a condensing engine.

Q. What arrangement of ports or passages is necessary to effect this result?

A. To effect this result there are needed three passages: one from the steam chest into each end of the cylinder, and one between them leading from the steam chest into the open air (or the condenser). (The steam chest ends of these *ports* must terminate in a flat surface. called the valve-

seat, on which the valve slides; and the ends of these passages or "ports" must be parallel with the steam and exhaust edges of the valve, and generally at right angles to the line of motion of the valve.

(In all cases where the term "slide valve" is used, it will be understood that the plain slide valve is meant, having two lips or faces and one arch, and its steam and exhaust edges parallel to each other, and at right angles to the line of motion.)

Q. How does the slide valve effect the alternate admission and exhaust of the steam from each end of the cylinder?

A. The arch of the valve is of sufficient span, and its legs so proportioned, as to enable it to make, during part of the movement, a communication between the central or exhaust port of the cylinder and one of the end ports (which do duty alternately as steam and exhaust passages), while during a portion of the movement the other end port is open to the ingress of the steam from the chest, which is in communication with the boiler.

Q. What is the simplest type of D valve?

A. The simplest type of D valve is that in which the arch has just enough span to

reach across the exhaust port and the two bridges or partitions, spaces between the steam and the exhaust ports, and the legs or lips are just as wide as the end or steam ports. (See Fig. 4.)

Q. What is the "travel," "throw" or "stroke" of a valve?

A. The travel, stroke or throw of a valve is the linear distance through which any part of it moves. In some cases it is fixed; in others variable. When not otherwise specified it is understood as invariable.

Q. How much travel should a D valve have?

A. It is generally best that a D valve have as much travel as will permit the end ports to be completely uncovered. If, however, this causes too much closing of the exhaust port by the exhaust edge of the valve the travel should be less, although this does not completely uncover the steam port. If the bridge is so narrow as to make it necessary to cramp either the admission or the exhaust, or both, the choice must be made with good judgment, based on principles to be laid down later.

Q. How is the D valve generally actuated?

A. The D valve is generally driven directly by an eccentric on the main shaft, although sometimes a rock-arm intervenes between the eccentric rod and the valve stem.

Q. How else may the valve be moved?

A. The valve may be moved by a crank, which is in many senses a mechanical equivalent of the eccentric, or by a yoke and block; or by tappets on the piston rod, or by a cam, or by steam.

[Where not otherwise specified, the motion is supposed to be by an eccentric, without any rock-arm. Where a rock-arm is referred to it will be supposed to be equal armed, unless the contrary is stated.]

Q. What is a lapless valve?

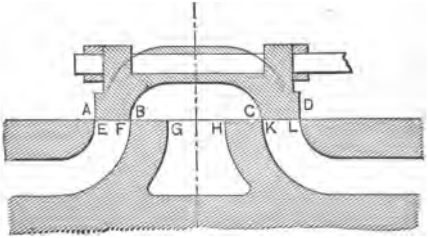


FIG. 4.—LAPLESS VALVE, CENTRAL POSITION.

A. A lapless valve, or valve without lap, is one in which the arch equals the exhaust port plus the bridges, and the legs or lips equal the end ports. (See Figs. 4, 7 and 9.)

Q. When is a valve said to have outside lap?

A. A valve is said to have outside lap

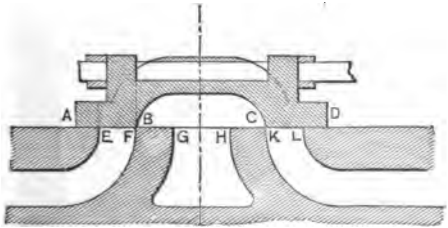


FIG. 5.—VALVE WITH ONLY OUTSIDE LAP. CENTRAL POSITION.

when its legs or lips project beyond the end ports when it is in mid position. (See Figs. 5, 6, 8, 10.)

Q. When has a valve inside lap?

A. A valve has inside lap when its arch is shorter than the sum of the exhaust port

and the bridges; in which case the exhaust

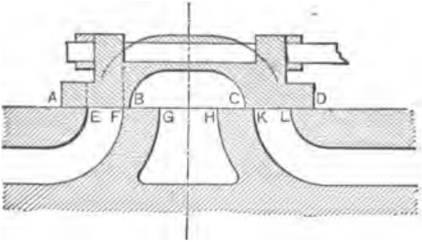


FIG. 6.— VALVE WITH BOTH OUTSIDE AND INSIDE LAP. CENTRAL POSITION.

edges lap over the outer edge port of the bridges. (See Figs. 6, 9, 10.)

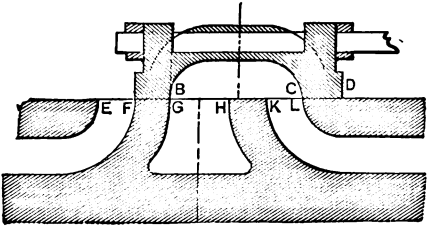


FIG. 7.—LAPLESS VALVE. FULL TRAVEL.

Q. How much "seal" should be allowed?

A. The amount of seal to allow is entirely

a matter of judgment, depending largely upon the straightness and sharpness of the

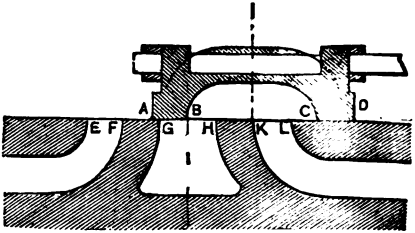


FIG. 8.—LAPLESS VALVE. OVER TRAVEL.

edges of port and valve, and the general condition of the surfaces in contact.

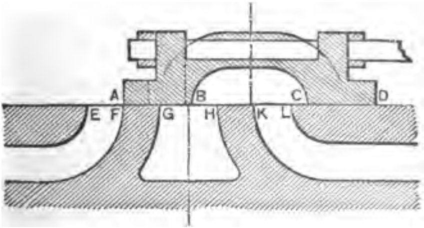


FIG. 9.—LAPPED VALVE. FULL TRAVEL.

Q. Suppose two valve-seats having equal steam ports and bridges, but one having

wider exhaust port than the other; what difference will be required in the valves?

A. If two valve-seats have the same bridges and steam ports, the one having the wider exhaust port will require a correspondingly wide arch. There will be no difference in travel, lap or point of cut-off.

Q. Suppose two valve-seats having equal steam ports and exhaust port, but different bridge-width; what will be the difference in the valve and its action?

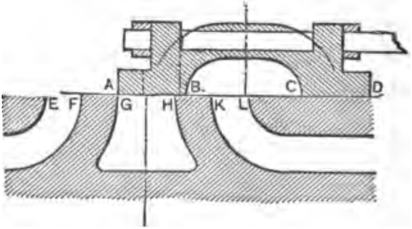


FIG. 10.—LAPPED VALVE. OVER TRAVEL.

A. If the steam and exhaust ports are the same in both seats, the one having the wider bridges will require proportionately wider arch. There will be no difference in the lap or the point of cut-off; but the one on the wide bridge may have the longer

travel of the two, and this will give free admission.

Q. If two valve-seats have the same exhaust port and bridges, but different width of steam port, what difference need there be in the valves?

A. If the exhaust port and the bridges are the same in both seats, then the one having the wider steam ports will require proportionately wider lips.

Q. What is steam lead?

A. Steam lead, generally known simply as "lead," refers to the *position* of the valve with reference to the piston, and not to any

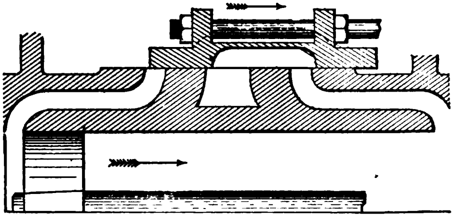


FIG. 11.—LAPPED VALVE, WITH STEAM LEAD.

dimension of the valve. It is a term referring to time and position, not to size and shape. A valve has lead when it is con-

stantly a little ahead of the position it would occupy if the steam edge was exactly in line with the outer edge of the end port. (See Fig. 11.)

Q. What is "negative steam" or "negative lead?"

A. "Negative steam lead," or simply "negative lead" is a contradictory term, (somewhat as though we spoke of "black whitewash") meaning that instead of the valve commencing to open for steam admission before the piston has reached stroke-end, it does not commence to open until the piston has commenced the stroke.

Q. How is it that with steam lead the piston can reach the end of its stroke when the valve has commenced to let steam in against it?

A. When a valve has moderate steam lead, the amount of opening before stroke end is very slight; just enough to clear the "seal" of the steam edge and give a slight crack of opening; hence the amount of steam thus admitted against the advancing face of the piston is very slight; not sufficient to overcome the momentum of the piston rod and crosshead, etc., to say nothing of the momentum of the fly wheel.

Q. When a valve has "*negative* steam lead," what starts the piston on its stroke before there is any steam to drive it?

A. When a valve has negative lead, of course the inertia of the reciprocating parts would tend to hold the piston at stroke end and prevent a new stroke being commenced; but here the momentum of the rotating fly wheel comes into play and the reciprocating parts are reversed and dragged along to commence the new stroke, until the valve opens to let steam against the piston-face.

Q. Is it essential that the lead shall be the same at both ends of the stroke?

A. *Usually* it is highly desirable that the lead shall be the same at both ends of the stroke?

Q. What is exhaust release?

A. Exhaust release, or simply "release," is opening the port for exhaust.

Q. What is cutting off?

A. Cutting off is closing the port between the cylinder and the steam chest, so as to use the expansive pressure of the steam during part of the stroke, instead of having steam chest pressure throughout.

Q. What is "cushioning" or "compression?"

A. Cushioning or compression is closing the end port between the cylinder and the exhaust port, so that the unexhausted steam shall be compressed by the advancing piston, and its pressure increased.

Q. What is the position of a lapless slide valve when the piston is at the end of its stroke supposing no lead?

A. A lapless valve, if it has no lead, is exactly at its central or neutral position when the piston is at either end of its stroke. (See Fig. 12.)

Q. In which direction does a lapless valve (without lead) move from its central position

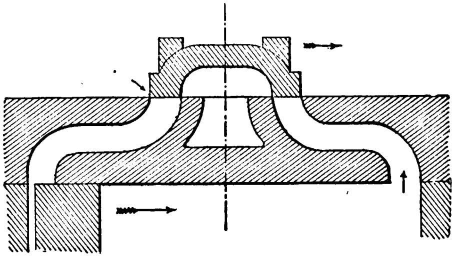


FIG. 12.—LAPLESS VALVE WITHOUT LEAD.

when the piston moves from the end of the cylinder?

A. A lapless valve (without lead) starting from its central position (see Fig. 12) moves in the same direction as the piston until the piston is at mid-stroke, and then moves in the opposite direction while the piston finishes its stroke.. In other words, the valve makes half a single stroke forwards and half a single stroke backwards, while the piston makes a whole single stroke forwards; and *vice versâ.*

Q. What is the position of a lapless valve (without lead) when the piston is at mid stroke?

A. When the piston is at mid-stroke a lapless valve having no lead is at the end of its travel.

Q. With a lapless valve what is the condition of the ports when the piston is at the back end of its stroke?

A. With a lapless valve, when the piston is at back stroke, the back port is just about to open into the chest as a steam passage, and the front port is acting as an exhaust passage, communicating through the arch of the valve with the central exhaust port, and through it with the open air (or with the condenser). (See Fig. 12.)

Q. When a valve has steam lap, what is

its position when the piston is at back stroke?

A. When a valve has steam lap it is, when the piston is at back stroke, exactly or a trifle more than the length of its lap ahead of its neutral position. (See Fig. 11.)

Q. When a valve has steam lap what is the condition of the ports when the piston is at back stroke?

A. With a lapped valve the ports are in the same condition as with one without lap.) (See Fig. 11.)

Q. How should the eccentric be set, where the valve has neither outside lap nor lead?

A. Where the valve has neither lead nor outside lap, the eccentric should be set 90 deg. ahead of the crank if the engine is to run forward; 90 deg. behind the crank if the engine is to run backward. (See Fig. 13.)

(Where there is a rock arm the relative positions should be reversed; that is, for a forward running engine the crank should be 90 deg. ahead of the eccentric, and if it is to run backward the crank should be 90 deg. behind the eccentric.)

Q. Where there is steam lap and no lead how should the eccentric be set?

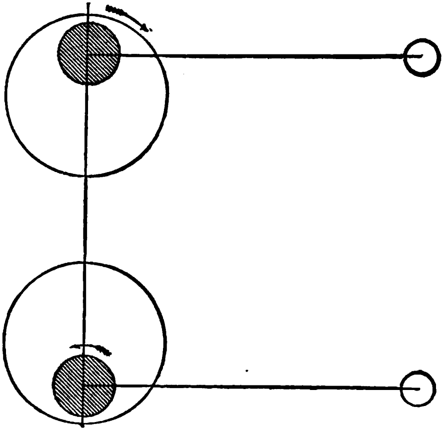


FIG. 13.—POSITIONS OF ECCENTRIC.

A. Where there is steam lap and no lead the eccentric should be set so that the back steam edge of the valve lines with the back edge of the back end port, when the piston is at back stroke; that is, when the crank is on the back centre.

Q. When the engine is to have steam lead where should the eccentric be set?

A. Where the engine is to have steam lead the eccentric should be set so that the back steam edge of the valve shall be in line with the back edge of the back port when the crank is just coming to its back centre; that is, below if it is an over-running engine, and above it runs "under" or backwards.

Q. When should the valve be set?

A. The valve should be set when it and the chest and all parts of the engine are hot.

Q. What alterations may be made in existing slide valves?

A. In existing slide valves the lead may be altered, inside lap added or removed, and outside lap added or removed; in some cases the travel may be increased or lessened.

Q. How is lap best added?

A. Lap is best added by attaching truly planed metal strips by means of flush screws; having previously trued the edges of the valve so that the new strips and the old valve-edges shall make a tight steam joint.

Q. How is the lap best removed when excessive?

A. Inside lap is best removed by careful chipping to line and dressing with a file.

Q. How is excessive outside lap best taken off?

A. Outside lap is best taken off by planing.

Q. How may valve seats be altered?

A. Valve seats may be altered by widening or narrowing either the exhaust or the end ports, by chipping away or adding to the outer edges of either of them, or by doing the same thing to the inner edges of the end ports, or by a combination of the above. In some cases the old seat is planed or chipped down sufficiently to allow of a new or "false" seat having the desired ports to be fitted. This is frequently done in locomotive practice.

Q. What might be given as a rule for proportionate width of steam and exhaust ports and bridges?

A. We might say for steam ports and bridges each 1 in., and for exhaust port, 1½ to 2 ins.

Q. What is the least travel of a lapped valve that will give full opening of the steam port?

A. The least travel of a lapped valve that

will give full opening of the steam port is equal to twice the lap plus twice the steam port width.

Thus: steam port, $1\frac{1}{4}''$, lap, $\frac{3}{4}''$, least travel, $2\frac{1}{2} + 1\frac{1}{2} = 4''$.

Q. How much arch or span should the valve have?

A. If there is neither inside lap nor "inside clearance" (negative inside lap) the arch should be equal to the sum of the two bridges and the exhaust port.

Thus; exhaust port $2''$, bridges $1''$ each, then arch should be $2'' + 2'' = 4''$, less a small amount for seal.

Q. To keep the steam port full open during any period of the travel, what should be done?

A. To keep the steam port full open during any part of the stroke, make the travel greater than double the sum of the outside lap and steam port *bridge*.

Q. How does increase of travel affect the compression?

A. Increase of travel retards compression.

Q. What is the effect on the exhaust release of increasing travel?

A. Greater travel gives later release.

Q. How does increased travel change the time of pre-admission, with a given lineal valve lead?

A. Keeping the lineal valve lead the same, increased travel delays pre-admission.

Q. What relation between travel and freedom of steam admission?

A. More travel, freer steam admission up to the point where overtravel chokes the exhaust.

Q. What effect has inside lap upon the time of admission?

A. Inside lap has no influence on steam admission.

Q. How does inside lap affect the point of cut-off?

A. Inside lap does not in any way affect the point of cut-off.

Q. Does inside lap increase or diminish compression?

A. Inside lap hastens compression, hence increases it.

Q. Does inside lap hasten or retard exhaust release?

A. Inside lap retards exhaust release.

Q. By what other name is inside lap known?

A. Inside lap is often known as "exhaust lap," and sometimes as "exhaust cover," and "inside cover."

Q. What effect has inside lap on the freeness of admission of steam?

A. Inside lap makes no difference in the steam admission.

Q. What effect has inside lap on freeness of exhaust?

A. Inside lap tends to choke the exhaust.

Q. What effect has inside lap on the expansion?

A. Inside lap allows expansion to take place during a longer period than if there were none.

Q. What effect has increase of outside lap on the cut-off and expansion?

A. Increasing outside lap hastens cut-off and prolongs expansion.

Q. Does increasing outside lap hasten or retard compression?

A. Increasing outside lap has no effect on compression.

Q. What effect has increasing outside lap, on exhaust release?

A. Increase of outside lap neither hastens nor retards exhaust release.

Q. How does increase of outside lap

affect pre-admission where the eccentric positions remain unchanged?

A. The more outside lap, the later admission takes place.

Q. How may retardation of cut-off, caused by increase of travel, be counteracted?

A. If the cut-off is made later, by reason of increased travel, increased outside lap will neutralize this.

Q. How may retardation of compression, due to increase of travel, be neutralized?

A. Delayed compression, caused by increased travel, may be corrected by more inside lap.

Q. How may too late release, due to too much travel, be remedied?

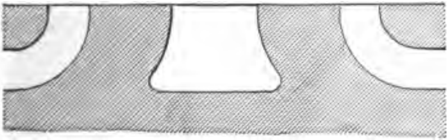


FIG. 14.—WIDE BRIDGES.

A. If the exhaust takes place too late, by reason of too much travel, the inside lap

may be cut out, if there is any, or if there is none, "inside clearance" may be given.

Q. What effect has "inside clearance," or "negative inside lap" on steam admission and on point of cut-off?

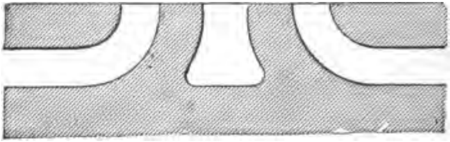


FIG. 15.—NARROW BRIDGES, NARROW EXHAUST.

A. "Negative inside lap" has no influence on either the time of admission or the point of cut-off.

Q. What effect has "negative inside lap" on compression?

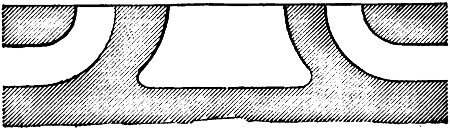


FIG. 16.—NARROW BRIDGES, WIDE EXHAUST.

A. Negative inside lap delays compression.

Q. How does negative inside lap affect exhaust release?

A. Negative exhaust lap hastens exhaust release.

Q. What effect has increasing the lead, upon the various operations of the valve?

A. Giving more lead hastens every operation of the valve.

Q. What is "seal?"

A. "Seal" is a slight overlapping of the steam edges of a valve to prevent leakage.

Q. How is "seal" best obtained?

A. "Seal" is best obtained by giving the valve less travel than the sum of the steam-ports and outside laps.

Q. What relation between the lead required and the speed?

A. The higher the speed the more lead required to cause smooth running.

Q. Suppose there is no outside lap?

A. If there is no outside lap then there is no cut-off and no expansion.

Q. Suppose there is no inside lap?

A. If there is no inside lap compression at one end and release at the other are simultaneous.

Q. If there is neither lap nor lead where should the eccentric be placed?

A. Where there is neither lap nor lead, the eccentric should be at right angles to the crank and ahead of it.

Q. If there is lap or lead where should the eccentric be?

A. Where there is lap or lead, the eccentric should be enough more than 90 deg. ahead of the crank to move the valve the length of the lap or lead. This increased angle of eccentric is called the "angular advance," and is counted from the 90 deg. as zero—*i.e.*, an angular advance of 30 deg. equals a total angle of 120 deg. of the eccentric line ahead of cranks.

Q. How may the evil effect of inside clearance, connecting the opposite ends of the cylinder, be counteracted?

A. The evil effect of inside clearance, in connecting the opposite ends of the cylinder, may be prevented or cured by adding an equal amount to the steam-edge of the valve-lip.

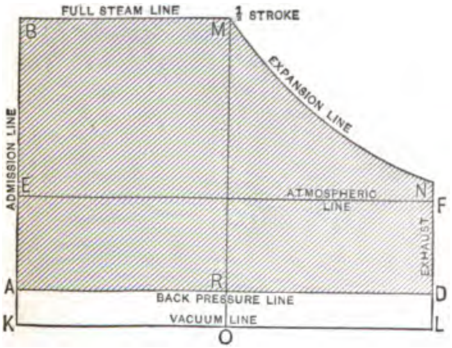
EXPANSION.

Q. What is working steam expansively?

A. "Working steam expansively is letting it into the working cylinder and cutting it

off before the piston has completed its stroke; making less than a cylinder full of steam, by its expansive force, do more work than the same weight of steam would do if allowed to follow the piston under full initial pressure." *

Q. If steam is cut off at half stroke of the piston, to what volume will it expand,



and what will be its pressure at the end of the stroke?

* "Miller, Millwright and Millfurnisher," by same author, p. 158.

A. If the steam is cut off at $\frac{1}{2}$ stroke, it tends to expand to $\frac{1}{2}$ absolute pressure and double volume. Its expansion is said to be twofold, or it is said to have an expansion rate of two. (*This is supposing perfect expansion, neglecting clearance, internal condensation, and other influences which exist in practice.*)

Q. What is the pressure of the steam at $\frac{3}{4}$ stroke, when it is cut off at $\frac{1}{2}$?

A. Where the steam is cut off at $\frac{1}{2} = \frac{2}{4}$ stroke, it will at $\frac{3}{4}$ stroke have expanded $\frac{3}{2} = 1.5$ fold; and its pressure will be $\frac{2}{3}$ that at the point of cut off.

Q. What is the average pressure of the steam in the cylinder, with twofold expansion?

A. With twofold expansion the average pressure above vacuum will be about

$$\frac{1.69}{2} = 84\frac{1}{2}$$

per cent. of the initial pressure above vacuum.

Q. With cut off at $\frac{1}{2}$ stroke, and twofold expansion, what is the gain in power of the engine?

A. With twofold expansion, cutting off at $\frac{1}{2}$, there is *no* "gain in power," or in capacity of the engine. The "power" or "capacity" at that speed and initial pressure is lessened. With one-half the steam, a quantity of work equal to only

$$\frac{1.69}{2} = 84\tfrac{1}{2}$$

per cent. of that done at full steam, is done; but the gain in the work got out of a *given weight* of steam is 69 per cent.

Q. How may the power of the engine working until cut off at $\frac{1}{2}$, be kept up to that when working without cut off?

A. The "capacity" or "power" of the engine working at $\frac{1}{2}$ cut off, may be kept the same as when working with full steam, by increasing either the initial pressure above vacuum or the speed to

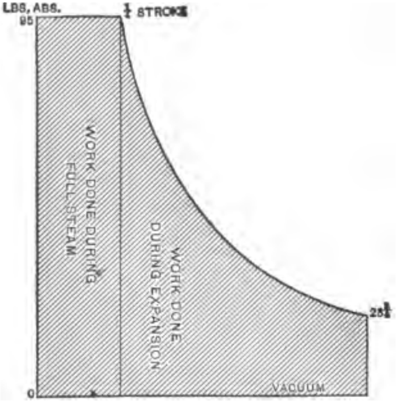
$$\frac{100}{84.5} = 1.183$$

times what it was when using full steam. For instance: if we have an engine making 80 revolutions per minute, and giving out 75 horse-power at 60 lbs., absolute pressure (above vacuum) at full stroke, and

we change it so as to cut off at half stroke, we would get only $75 \times .845 \times 63.4$ horse-power; and we would have to raise the initial pressure above vacuum to

$$\frac{60 \times 75}{63.4} = 70.9 \text{ lbs. } (= 56.2 \text{ lbs. by gauge})$$

or raise the speed to $80 \div .845$ or $80 \times 1.183 = 94.64$ turns per minute, in order to get the original 75 horse-power.



Q. With steam cut off at ¼ stroke, and fourfold expansion, what should be the pres-

sure above vacuum, and the volume, at the end of the stroke?

A. Cutting off at ¼, the terminal pressure above vacuum should be ¼ the initial pressure above vacuum, as the volume would be fourfold. (This is supposing that there was no clearance.)

Q. Cutting off at ¼, what would be the volume and absolute pressure at ½ stroke?

A. Cutting off at ¼, the volume and half stroke would be doubled, and the absolute pressure reduced to ½ what it was at cut off.

Q. Cutting off at ¼, what would be the absolute pressure at ¾ stroke?

A. With cut off at ¼, the volume at ¾ stroke would be trebled, and the absolute pressure reduced to only ⅓ that at cut off.

Q. Cutting off at ¼, and expanding fourfold, what is the proportion between the work done during expansion, and that done during full steam?

A. Cutting off at ¼ and getting four-fold expansion, the work done during the last ¾ stroke, while expansion is taking place, is measured by the "hyperbolic logarithm" of four, the expansion rate; this hyper-

bolic logarithm of four being 1.3863. In other words, if the engine gave out 400 horse-power with full steam, it should, with cut off at $\frac{1}{4}$, give out 100 horse-power during the $\frac{1}{4}$ period of full steam, and 138.63 horse-power during expansion ; total, 238.6 horse-power, with $\frac{1}{4}$ the steam required to to give out 400 horse-power.

Q. Cutting off at $\frac{1}{4}$, and expanding four-fold, what should be the average pressure above vacuum during the whole stroke?

A. Cutting off at $\frac{1}{4}$ and expanding four-fold, the average pressure above vacuum *during the whole stroke,* is calculated by dividing the expansion rate into one plus, the hyperbolic logarithm of the expansion rate; thus the average pressure above vacuum in this case would be

$$\frac{2.386}{4} = .5965 \text{ times the initial}$$

pressure above vacuum.

Q. Cutting off at $\frac{1}{4}$, and expanding four-fold, what should be the *average* proportionate pressure above vacuum *during expansion?*

A. Cutting off at $\frac{1}{4}$, and expanding four-

fold, the *average* proportionate pressure above vacuum *during expansion* is best obtained by dividing the hyperbolic logarithm of the expansion rate by the ratio between the "full steam," and the "expansion" portions of the stroke. Thus, the expansion part of the stroke in this case, is three times the full steam portion; and the average pressure *during expansion* should be

$$\frac{1.386}{3} = .462 \text{ times the initial pressure.}$$

(This calculation, however, is of little or no use in practice.)

It must be distinctly understood (1) that pressures must be reckoned above vacuum in calculating expansion; (2) that the actual expansion rate cannot be obtained without knowing not only the clearance between the piston head and the cylinder head at stroke end (when the engine is on the dead center) but the waste space in the steam passage, between the cylinder and the valve-face; (3) that the mean effective pressures are got by subtracting the mean back pressures above vacuum from the mean total pressures above vacuum.

The following table gives the mean total pressures with eight different actual expansion rates, and at initial pressures ranging from 25 to 140 lbs. per square inch, reckoning above vacuum.

MEAN TOTAL PRESSURES OF EXPANDING STEAM.

Initial Pressure above Vacuum.	*Actual Expansion Rates.*			
	1.333	1.5	1.625	2.
25	24.130	23.481	22.938	21.164
30	28.956	28.100	27.524	25.396
35	33.782	32.784	32.110	29.630
40	38.608	37.468	36.700	33.862
45	43.434	42.215	41.288	38.095
50	48.262	46.835	45.875	42.328
55	53.088	51.518	50.462	46.561
60	57.914	56.202	55.050	50.794
65	62.740	60.885	59.637	55.027
70	67.566	65.569	64.225	59.260
75	72.393	70.252	68.812	63.493
80	77.216	74.936	73.400	67.726
85	82.012	79.619	77.987	71.959
90	86.866	85.303	82.574	76.192
95	91.699	89.986	87.163	80.425
100	96.524	93.670	91.750	84.657
105	101.35	98.353	96.337	88.890
110	106.17	103.04	100.92	93.123
115	111.00	107.72	105.51	97.356
120	115.83	112.40	110.10	101.59
125	120.65	117.08	114.68	105.82
130	125.48	121.77	119.27	110.05
135	130.30	126.45	123.86	114.28
140	135.13	131.13	128.45	118.52

MEAN TOTAL PRESSURES OF EXPANDING STEAM.—*Con.*

Initial Pressure above Vacuum.	*Actual Expansion Rates.*			
	2.666	3	4	8
25	18.567	17.488	14.913	9.6232
30	22.280	20.986	17.897	11.548
35	25.992	24.484	20.880	13.472
40	29.706	27.982	23.863	15.397
45	33.420	31.479	26.844	17.321
50	37,133	34.977	29.828	19.246
55	40.846	38.474	32.811	21.170
60	44.459	41.972	35.794	23.095
65	48.273	45.470	38.777	25.020
70	51.986	48.967	41.760	26.944
75	55.700	52.465	44.743	28.869
80	59.413	55.963	47.726	30.794
85	63.126	59.461	50.709	32.718
90	66.840	62.958	53.692	34.643
95	70.553	66.456	56.675	36.568
100	74.267	69.954	59.657	38.493
105	77.981	73.451	62.640	40.417
110	81.694	76.949	65.622	40.342
115	85.407	80.477	68.606	44.267
120	89.121	83.944	71.589	46.191
125	92.834	87.442	74.572	48.116
130	96.548	90.940	77.555	50.041
135	100.26	94.437	80.538	51.966
140	103.97	97.935	83.520	53.890

The foregoing table is prepared without any consideration of clearance, and is merely a somewhat close approximation, of use for engines having but slight clearance—say 2 per cent.

Q. What is "initial expansion?"

A. Initial expansion is the expansion in volume which takes place during steam admission and before cut off.

Q. Is initial expansion desirable?

A. Initial expansion is desirable in a throttling engine; and sometimes when the steam is wet, with an automatic cut off.

Q. With two-fold expansion cutting off at one-half, how much gain of power is there by expansion, that is, how much work is done during expansion as compared with that done at full pressure?

A. When the cut off is at one-half and the expansion two fold, the work done during expansion is 0.69 that during full steam, and the gain in work done per pound of steam is, therefore 69 per cent.

Q. Under what conditions do clearance and waste volume have the most influence?

A. The shorter the stroke and the earlier the cut off, the greater the influence of a given clearance and waste volume (not proportion) in the expansion rate; and the earlier the cut off the greater the influence of a given proportion of clearance and

waste volume compared with the piston-displacement.

Thus the illustration given on page 29 shows the ill effects of too much clearance even where there is plenty of cushion.

The card with 5 per cent. clearance has the higher terminal pressure, but there is more than enough excess of loss of power by cushion to the same amount, to counterbalance the greater terminal pressure. The excess of the card having full expansion line in that part bounded by the expansion line, is more than made up by its deficit at the "cushion end" of the card.

Q. What is actual expansion ratio?

A. Actual expansion ratio takes into account the influence of clearance in the cylinder and in the steam passages, in lessening the nominal expansion rate. Thus, with cut off at ¼ stroke and clearance spaces at each end equal to 2% of the piston displacement, the actual expansion rate is only

$$\frac{1. + .02}{.25 + .02} = 3.78$$

The accompanying table, published only in part before, was calculated for the

author's use. It shows the actual expansion rates (neglecting piston rod area throttling, wire-drawing, internal condensation, condensation from radiation, &c.,) at various cut off rates and for various percentages of clearance and waste space.

It would be utterly impossible to construct a table large enough to include every combination of clearance space and point of cut-off found in actual practice. The author recommends that the reader (to see that he has the rule correctly) calculate for himself the actual expansion rates corresponding to the clearance and the point of cut-off nearest those of his own engine; and that then, after learning the clearance of his own engine exactly, he calculate the actual expansion rate for the full range of points of cut-off possible for that engine, advancing by hundredths of the stroke.

TABLE OF ACTUAL EXPANSION RATES.

Per Cent. of Clearance.	POINTS OF CUT OFF.							
	.10	.125	.20	.25	.30	.333	.375	.40
.01	9.181	7.481	4.809	3.884	3.258	2.944	2.628	2.463
.0125	9.	7.363	4.764	3.875	3.24	2.930	2.612	2.454
.0150	8.826	7.25	4.720	3.830	3.222	2.916	2.60[illegible]	2.445
.0175	8.659	7.133	4.677	3.803	3.204	2.902	2.592	2.436
.02	8.5	7.034	4.635	3.777	3.187	2.889	2.582	2.428
.0225	8.346	6.932	4.595	3.752	3.170	2.876	2.574	2.420
.0250	8.2	6.833	4.555	3.727	3.153	2.863	2.562	2.411
.0275	8.088	6.738	4.516	3.702	3.137	2.850	2.552	2.403
.03	7.933	6.645	4.417	3 678	3.121	2.837	2.543	2.395
.0325	7.792	6.555	4.440	3.654	3.105	2.824	2.533	2.387
.0350	7.666	6.468	4.404	3.631	3.089	2.812	2.524	2.379
0375	7.545	6.390	4.484	3.608	3.074	2.800	2.515	2.371
.04	7.428	6.303	4.333	3.58	3.058	2.788	2.506	2.363
.0425	7.315	6.229	4.298	3.564	3.043	2 776	2.497	2.355
.0450	7.206	6.147	4.256	3.542	3.028	2.764	2.488	2.348
.0475	7.102	6.082	4.232	3.521	3.014	2.752	2.479	2.340
.05	7.	6.	4.2	3.5	3.	2.741	2.470	2.333
.0525	6.901	5.985	4.163	3.478	2.986	2.730	2.461	2.325
.0550	6.806	5.861	4.130	3.459	2.971	2.719	2.453	2.318
.0575	6.714	5.794	4.106	3.439	2.957	2.708	2.445	2.311
.06	6.625	5.729	4.076	3.418	2.944	2.697	2.436	2.304
.0625	6.538	5.606	4.047	3.407	2.931	2.686	2.428	2.297
.0650	6.454	5.605	4.045	3.380	2.917	2.675	2.420	2.290
.0675	6.373	5.545	3.990	3.362	2.904	2.665	2.412	2.283
.07	6.294	5.482	3.963	3.342	2.892	2.655	2.404	2.276

TABLE OF ACTUAL EXPANSION RATES.

Per Cent. of Clearance.	POINTS OF CUT OFF.							
	.50	.60	.625	.70	.75	.80	.875	.90
.01	1.983	1.655	1.590	1.422	1.328	1.246	1.141	1.109
.0125	1.975	1.653	1.588	1.421	1.327	1.246	1.140	1.109
.0150	1 970	1.650	1.585	1.419	1.326	1.245	1.140	1.109
.0175	1.966	1.647	1.583	1.418	1.325	1.244	1.140	1.108
.02	1.961	1.645	1 581	1.416	1.325	1.243	1.138	1.108
.0225	1.956	1.642	1.579	1.415	1.324	1.243	1 138	1.108
.0250	1.952	1.640	1.576	1.413	1.322	1.242	1.138	1.108
.0275	1.947	1.637	1.574	1.412	1.321	1.241	1.138	1.107
.03	1.943	1.634	1.572	1.410	1.320	1.240	1.138	1.107
0325	1.938	1.632	1.570	1.409	1.319	1.240	1.138	1.107
.0350	1.934	1.629	1.568	1.408	1.318	1.239	1.137	1.106
.0375	1.930	1.627	1.566	1.406	1 317	1.238	1.136	1.106
.04	1.925	1.625	1.563	1.405	1.316	1.238	1.136	1.106
.0425	1.921	1.622	1.561	1.404	1.315	1.237	1.136	1.106
.0450	1.917	1.620	1.569	1.402	1.314	1.236	1.135	1.105
.0475	1.913	1.617	1.557	1.401	1.313	1.235	1.135	1.105
.05	1.907	1.615	1.555	1.400	1.312	1.235	1 135	1.105
.0525	1.904	1.613	1.553	1.398	1.311	1.234	1.134	1.104
.0550	1.900	1.610	1.551	1.397	1 310	1.233	1.134	1.104
.0575	1.896	1.608	1.549	1.396	1.309	1.233	1.134	1.104
.06	1.892	1.606	1.547	1.394	1.308	1.232	1.133	1.104
.0625	1.888	1.603	1.545	1 393	1.307	1.231	1.133	1.103
.0650	1.884	1.601	1.543	1.392	1.306	1.231	1.132	1.103
.0675	1.881	1.599	1.541	1.390	1.305	1.230	1.132	1.103
.07	1.877	1.597	1.539	1.389	1.304	1.229	1.132	1.103

Q. What is nominal expansion ratio?

A. Nominal expansion ratio is the proportion between the full stroke of the piston and that portion of the stroke during which the steam is not cut off. Thus, where the steam is cut off at $\frac{1}{4}$ stroke, the nominal expansion rate is 4.

Q. How early may a slide valve cut off, and why?

A. The ordinary D valve cannot cut off earlier than $\frac{5}{8}$ stroke without cramping the exhaust (unless it has "negative lead,") and $\frac{3}{4}$ is a more common point.

Q. What is the maximum grade of expansion with an ordinary slide-valve engine?

A. The maximum grade of expansion, allowing the cut off to be at $\frac{5}{8} = .625$ stroke, would be $\frac{8}{5} = 1.6$, without allowing for clearance; and allowing 5 % for clearance at each end, this is reduced to

$$\frac{1. + .05}{.625 + .05} = 1.5555$$

Q. How late can the Corliss type of automatic engines cut off?

A. Some of the Corliss types cannot cut off later than $\frac{1}{2}$: others as late as $\frac{3}{4}$.

Q. Why is it that some automatic engines can cut off later than $\frac{5}{8}$?

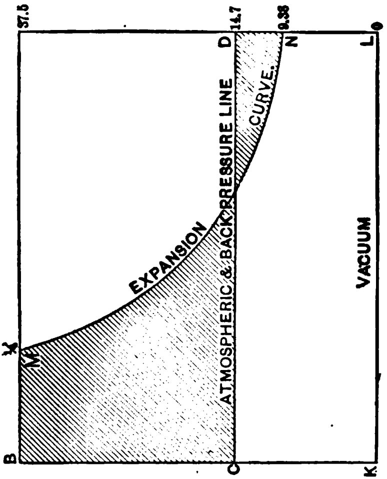
A. As a general thing, automatic engines have a fixed cut-off at some definite maximum point; and if earlier cut-off be desired, it is effected by an auxiliary (or special cut-off) valve, or by a device which closes the admission earlier than the regular positive motion would affect it.

In some engines (having only a single eccentric releasing gear) the governor can control the time of release only during the opening of the valve (necessarily during the first half of the stroke), and if the cut-off is not effected before mid-stroke it will not occur at all during that stroke.

Q. When the steam is expanded below the atmospheric pressure, in a non-condensing engine, what is the result?

A. When the steam is exhausted below the atmospheric pressure in a non-condensing engine, there is a return of steam from the exhaust passages into the cylinder. This condition of affairs is shown in the

figure (in which please note that there is no allowance for clearance).



(In all the foregoing there is no allowance made for compression.)

THE CONDENSER.

Q. What is a jet condenser?

A. A jet condenser is one in which the exhaust steam is condensed by actual con-

tact with a jet or spray of cold water injected by a pump or by other means.

Q. What is a surface condenser?

A. A surface condenser is one in which the exhaust steam is condensed by contact with the walls of tubes or sheets, kept cold by a constant circulation of water.

Q. How much improvement ought to be made in the duty by adding a condenser?

A. A good condenser will add for stationary about 10 lbs. and for marine 14 lbs. to the mean effective pressure, with the same terminal pressure, or will give the same mean effective pressure with correspondingly less terminal pressure.

Q. What is the advantage of the surface condenser?

A. The surface condenser allows the same feed water to be used over and over again, where pure or soft water is scarce.

Q. What is the objection to surface condensers?

A. The objection to surface condensers is that they are apt to cause corrosion of the boilers—especially where animal oils are used for steam chest and cylinder lubrication.

Q. Should the jet condenser be kept at

a low temperature when the feed water is taken from it?

A. Where the feed is taken from the condenser overflow it may best have its temperature raised by passing through a coil where it will be further heated by the gases of combustion.

Q. How much injection water will be required in a jet condenser to maintain a good vacuum and not let the temperature get too high?

A. About 25 times the weight of the feed water may be counted on.

Q. Where is the gain in the use of a condenser most marked?

A. The gain in the use of a condenser is most marked when the piston area is large.

Q. What should be the temperature of the condensation water with a jet cendenser?

A. The temperature of the condensation water with jet condensation should not exceed 120 deg. F (= 84½ deg. C.) and might very well be about 105 deg. F. (= 76 deg. C. The best results are obtained from a range between 108 deg. and 110 deg. F.

Q. How much heat will a pound of steam carry to the condenser?

A. If there is a back pressure of 10 lbs. absolute (4.7 lbs. vacuum) then there will be in it 1140.89 thermal units above 82 deg.; ifthe best well discharge is at 100 deg., the condenser takes 1040.89 thermal units.

Q. How much condensing water is required per hour per horse-power.

A. The amount of condensing water per hour per horse power varies. The colder the water the less required. Say exhaust steam is at 10 lbs. absolute back pressure and the injection water is at 60 deg. and hot well discharge 100 deg., then each pound of water absorbs 40 thermal units. Exhaust at 10 lbs. absolute has 1140.89 thermal units, then the condenser gets 1140.89 — 100 = 1040.89 thermal units per pound of steam, and there will be needed

$$\frac{1040.89}{40} = 26.022 \text{ lbs.}$$

of injection water for every pound of steam. If there were used 23 lbs. of steam per hour per horse-power, then there will be needed say 23 × 26.022 = about 600

lbs. of injection water per hour per horse-power, =

$$\frac{600}{60} \text{ or } 10 \text{ lbs.}$$

per minute per horse-power,

$$\frac{10}{8.5} \text{ or } 1\tfrac{1}{8} \text{ gallons per minute per horse-}$$

power.

With injection water at 80 deg. F, there would be needed

$$\frac{1040.89}{160^\circ - 80^\circ} = 52.045 \text{ lbs.}$$

of injection water for each pound of steam, or

$$\frac{52.045 \times 23}{60 \times 8.5} = 2.347 \text{ gals.}$$

per minute per horse-power.

Q. What is the simplest form of condenser?

A. The simplest form is the "siphon condenser."

Q. How much water does a siphon condenser require?

A. A Wheelock siphon condenser re-

quires about 1 gallon of condensing water per minute per horse-power.

Q. What is a desirable feature in a siphon condenser?

A. A desirable feature in a siphon condenser is adjustability of vacuum and capacity.

Q. Upon what does the gain of power by the use of a condenser depend?

A. The gain by using a condenser depends supon the back and mean effective pressures, prior to adding the condenser; upon the degree of vacuum obtained, and upon the amount of power required to operate the condenser.

Q. How does the load on the engine affect the proportionate gain by the condenser?

A. The greater the load the less the proportionate gain by condensing.

Q. How does the back pressure affect the proportion of gain by using a condenser?

A. The greater the back pressure before using a condenser, the greater the economy in its use.

Q. How much water is required to operate a condenser?

A. It is usually rated that a condenser requires one to two gallons per minute per horse-power.

Q. What effect has the temperature of the condensing water upon the capacity and economy of the condenser.

A. The colder the injection water, the greater the capacity and economy of a condenser, if the water supply is at all limited.

Q. Why are high steam pressures advantageous?

A. High steam pressures are advantageous because the proportion of heat required to raise water to steam at atmospheric pressure, compared with that required to bring it to working pressure, is less with high than with low pressures.

Q. What are the objections to very high initial pressure and early cut off?

A. The objections to very high initial pressure and early cut off are the shocks upon the moving parts, the decomposition of lubricants, increased leakages, and larger cost for cylinder, framing and foundation.

Q. Which are the more economical, large or small engines?

For the same *proportionate* load, large engines are the most economical. Thus, a 100-horse engine, cutting off at ¼, would be more economical than two 50's with the same expansion rates and general design.

Q. In a non-condensing engine, how may we get the best economy with a given boiler pressure?

A. We may get out of a non-condensing engine the best possible duty with a given boiler pressure, by keeping the full pressure clear up to the point of cut-off, and expanding down nearly to the atmospheric pressure (supposing free exhaust and admission, and minimum clearance, friction, leakage and condensation).

Q. What are the results of too light loads?

A. The results of too light loads (or too large engines) are excessive internal condensation, and in some cases expansion below the atmospheric pressure, making a partial vacuum on what should be the working side of the piston, dragging on the fly wheel.

Q. When is the best economy attained?

A. The best economy is considered by most engineers to be attained "when the mean effective pressure is highest relatively to the terminal pressure."

Q. What is the exact measure of the work done by an engine?

A. The exact measure of the work done is the mean effective pressure.

Q. What is the measure of the steam consumption of the engine?

A. The steam consumption may be measured by the terminal pressure.*

Q. What is the effect of early exhaust closure on steam consumption?

A. Early exhaust closure saves steam.

Q. What is the effect of exhausting from the clearance at a pressure greater than the back pressure?

A. Exhausting from the clearance at a pressure greater than the back pressure wastes steam.

Q. How may the water consumption of an engine be calculated?

A. The water consumption of an engine in pounds per hour, may be calculated by

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* See definitions of Pressures, page 21.

dividing 859,375 by the *volume* of steam at the terminal pressure, and by the mean effective pressure.

Q. How is the economy of a steam engine expressed?

A. The "economy" or "duty" of a steam engine ought to be expressed in pounds of water consumed per hour per horse power, and not in pounds of coal.

Q. Why should the economy be expressed in pounds of steam (or water) rather than in pounds of fuel per hour per horse-power?

A. The amount of coal or other fuel per hour used to generate steam for an engine of any stated horse-power depends on conditions entirely independent of the engine which uses the steam—such, for instance, as the kind, size, proportions and condition of boiler, the way it is set and fired, the kind and temperature of the feed water, the draft, the kind of fuel, the dryness of the steam, etc.

Q. Which are the most economical, high or low initial cylinder pressures; and why?

A. High pressures are the most economical, because the proportion of lost heat and

pressure in the exhaust to the total pressure is less.

Q. What are the advantages of dry steam?

A. Water in the steam not only lessens the capacity and duty of the engine, but is dangerous to the cylinder heads.

Q. How may the danger from water in the cylinder be lessened?

A. (1) By lagging the steam pipe; (2) by having the exhaust at the side or underneath; (3) by having a "pop-piece" in the head, much weaker than the rest of the head, and put in like a hand-hole plate, so that in case of over-pressure it will give way and relieve the pressure.

Q. What are the relative economies of steam at various grades of expansion?

A. The relative economies of steam at various grades of expansion are given in the accompany large table from D. K. Clark; the clearance being assumed at 7% at each end of the stroke, which corresponds to actual practise in a great many slide-valve engines:

Expansive Working of Steam: —Actual Ratios of Expansion, with the Relative Periods of Admission, Pressures, and Performance.

Clearance at each end of Cylinder, 7 per cent. of stroke.

(SINGLE CYLINDERS.)

1	2	3	4	5	6	7
Actual Ratio of Expansion; or Number of Volumes to which the Initial Volume is Expanded.	Hyperbolic Logarithm of Actual Ratio of Expansion.	Corresponding period of Admission, or Cut-off. Clearance, 7 per cent. of the stroke.	Average Total Pressure.	Total Initial Pressure.	Total Final Pressure.	Ratio of Total Performance of Equal Weights of Steam. (Col. 4 ÷ Col. 6)
Initial volume = 1.		Stroke = 100.	Initial pressure = 1.	Mean Pressure = 1.	Initial pressure = 1.	With 100 per ct. of admission = 1,000.
1.0	.0000	100	1.000	1.000	1.000	1.000
1.05	.0488	95.0	.9997	1.003	.952	1.050
1.1	.0953	90.3 or $\frac{9}{10}$	.996	1.004	.909	1.096
1.15	.1398	86.0	.990	1.010	.870	1.138
1.18	.1698	83.3 or $\frac{5}{6}$	.986	1.014	.847	1.164
1.2	.1823	82.1	.983	1.017	.833	1.180
1.23	.2070	80.0 or $\frac{4}{5}$	.980	1.020	.813	1.206
1.25	.2231	78.6	.977	1.024	.800	1.221
1.3	.2624	75.3 or $\frac{3}{4}$	.969	1.032	.769	1.261
1.35	.3000	72.3	.961	1.041	.741	1.297
1.39	.3293	70.0 or $\frac{7}{10}$	.953	1.049	.719	1.325
1.4	.3365	69.4	.951	1.052	.714	1.332
1.45	.3716	66.8 or $\frac{2}{3}$	.942	1.062	.690	1.365

1.5	.4055	64.3	.932	1.073	.666	1.399
1.54	.4317	62.5 or $\frac{5}{8}$	.925	1.081	.649	1.425
1.55	.4382	62.0	.922	1.085	.645	1.429
1.6	.4700	59.9 or $\frac{3}{5}$	.913	1.095	.625	1.461
1.65	.5008	57.9	.903	1.107	.606	1.490
1.7	.5306	56.0	.894	1.119	.588	1.520
1.75	.5595	54.1	.883	1.132	.571	1.546
1.8	.5878	52.4	.873	1.145	.555	1.573
1.85	.6153	50.8	.864	1.157	.541	1.597
1.88	.6314	50.0 or $\frac{1}{2}$	.860	1.163	.532	1.616
1.9	.6419	49.3	.854	1.171	.526	1.624
1.95	.6678	47.9	.846	1.182	.513	1.649
2.0	.6931	46.5	.836	1.196	.500	1.672
2.1	.7419	44.0	.818	1.222	.476	1.718
2.2	.7885	41.6	.799	1.251	.455	1.756
2.28	.8241	40.0 or $\frac{2}{5}$	.787	1.271	.439	1.793
2.3	.8329	39.5	.782	1.279	.435	1.798
2.4	.8755	37.6 or $\frac{3}{8}$	.766	1.305	.417	1.837
2.5	.9163	35.8	.750	1.333	.400	1.875
2.6	.9555	34.2	.736	1.359	.385	1.912
2.65	.9745	33.3 or $\frac{1}{3}$	.726	1.377	.377	1.925
2.7	.9933	32.6	.719	1.391	.370	1.943
2.8	1.030	31.2	.706	1.416	.357	1.978
2.9	1.065	29.9 or $\frac{3}{10}$	.692	1.445	.345	2.006
3.0	1.099	28.7	.679	1.473	.333	2.039
3.1	1.131	27.5	.665	1.504	.323	2.059
3.2	1.163	26.4	.652	1.534	.313	2.083
3.3	1.194	25.4	.641	1.560	.303	2.115
3.35	1.209	25.0 or $\frac{1}{4}$	.637	1.570	.298	2.129
3.4	1.224	24.5	.631	1.585	.294	2.146
3.5	1.253	23.6	.619	1.615	.286	2.164
3.6	1.281	22.7	.608	1.645	.278	2.187
3.7	1.308	21.9	.597	1.675	.270	2.211
3.8	1.335	21.2	.589	1.698	.263	2.240

Expansive Working of Steam:—Actual Ratios of Expansion, with the Relative Periods of Admission, Pressures, and Performance.

Clearance at each end of Cylinder, 7 per cent. of stroke.

(SINGLE CYLINDERS.)

1	2	3	4	5	6	7
Actual Ratio of Expansion, or Number of Volumes to which the Initial Volume is Expanded.	Hyperbolic Logarithm of Actual Ratio of Expansion.	Corresponding period of Admission or Cut-off. Clearance, 7 per cent. of the stroke.	Average Total Pressure.	Total Initial Pressure.	Total Final Pressure.	Ratio of Total Performance of Equal Weights of Steam. (Col. 4 ÷ Col. 6.)
Initial volume = 1.		Stroke = 100.	Initial pressure = 1.	Mean Pressure = 1.	Initial pressure = 1.	With 100 per ct. of admission = 1,000.
3.9	1.361	20.4	.579	1.727	.256	2.262
4.0	1.386	19.7 or $\frac{1}{5}$	.567	1.764	.250	2.278
4.1	1.411	19.1	.559	1.786	.244	2.291
4.2	1.435	18.5	.551	1.815	.238	2.315
4.3	1.459	17.9	.542	1.845	.233	2.326
4.4	1.482	17.3	.533	1.876	.227	2.348
4.5	1.504	16.8 or $\frac{1}{6}$	.526	1.901	.222	2.370
4.6	1.526	16.3	.518	1.930	.217	2.387
4.7	1.548	15.8	.511	1.957	.213	2.399
4.8	1.569	15.3	.503	1.988	.208	2.418
4.9	1.589	14.8	.494	2.024	.204	2.422
5.0	1.609	14.4 or $\frac{1}{7}$	.488	2.049	.200	2.440
5.2	1.649	13.6	.476	2.101	.193	2.466

5.4	1.686	12.8	.462	2.164	.185	2.497
5.5	1.705	12.5 or $\frac{1}{8}$	.457	2.188	.182	2.511
5.6	1.723	12.1	450	2.222	.178	2.528
5.8	1.758	11.4	.438	2.283	.172	2.547
5.9	1.775	11.1 or $\frac{1}{9}$	.432	2.315	.169	2.556
6.0	1.792	10.8	.427	2.342	.167	2.567
6.2	1.825	10.3	.419	2.387	.161	2.585
6.3	1.841	10.0 or $\frac{1}{10}$	.413	2.421	.159	2.597
6.4	1.856	9.7	.407	2.457	.156	2.609
6.6	1.887	9.2 or $\frac{1}{11}$	.398	2.513	.152	2.619
6.8	1.917	8.7	.388	2.577	.147	2.639
7.0	1.946	8.3 or $\frac{1}{12}$	.381	2.625	.143	2.664
7.2	1.974	7.9	.373	2.681	.139	2.683
7.3	1.988	7.7 or $\frac{1}{13}$	.369	2.710	.137	2.698
7.4	2.001	7.5	.365	2.740	.135	2.703
7.6	2.028	7.1 or $\frac{1}{14}$	.357	2.801	.132	2.711
7.8	2.054	6.7 or $\frac{1}{15}$	.348	2.874	.128	2.719
8.0	2.079	6.4 or $\frac{1}{16}$	.342	2.924	.125	2.736

Q. How has the economy of superheating steam been calculated?

A. Rankine calculates that in an engine taking steam at 34 gauge lbs., and expanding five-fold, there is 15% economy in superheating the steam from 257 deg. to 428 deg. F., not using waste heat to superheat with. Superheating with waste heat, the gain becomes 23%.

Q. Where is steam jacketing least effective, and why?

A. Steam jacketing is less effective in

cylinders of great diameter and quick stroke, as the heat from the jacket does not penetrate to the centre of the cylinder soon enough to prevent internal condensation.

Q. Can a steam jacket be an advantage without increasing the economy?

A. A steam jacket may, without saving any steam, increase the capacity of a small engine, by bringing the mean effective pressure higher.

Q. Where is steam jacketing effected?

A. Steam jacketing is most usually effected on sides and ends of cylinders, though sometimes it is omitted from the cylinder heads, and occasionally the pistons are made hollow and kept full of live steam.

Q. What are the relative theoretical economies of steam at various pressures in a non-condensing engine?

A. The following table gives the mean effective pressures, pounds of water per hour per horse-power, gain of power, and economy of fuel, for various initial cylinder pressures above atmosphere, with cut-off at $\frac{1}{6}$:—

TABLE SHOWING ADVANTAGES OF USING HIGH PRESSURES.

(Non-condensing engine cutting off at one-sixth stroke.)

INITIAL PRESSURE.	M. E. P. Lbs. per sq. in.	Lbs Water per Hour per H. P.	Gain of Power per Increment of 10 lbs.	Economy of Fuel per Increment of 10 lbs.
10 gauge lbs...	9.3	75.3	pr. ct.	pr. ct.
20 " "...	16.3	42.9	76	43
30 " "...	21.2	33.0	30	23
40 " "...	25.6	27.3	20	17
50 " "...	29.1	24.0	14	12
60 " "...	32.0	21.9	10	09
70 " "...	34.5	20.3	08	07
80 " "...	36.3	19.2	06	05
90 " "...	38.1	18.4	05	04
100 " "...	39.2	17.8	03	03

Q. What is superheated steam?

A. Referring to the definition of saturated steam: "When a body of saturated steam is isolated from water in a space of fixed dimensions, if an additional quantity of heat be supplied" to it, it "becomes superheated, and the temperature and pressure are increased, while the density

is not increased." Steam, thus surcharged with heat, approaches to the condition of a perfect gas.

Q. What is the advantage of superheating steam?

A. The advantage of superheating steam is to lessen internal condensation.

Q. What is the disadvantage of superheating steam?

A. The disadvantages of superheated steam are that it is more dangerous, requires greater caution in handling, and causes excessive leakage, wear and tear.

Q. Is a steam jacket advantageous?

A. The steam jacket is advantageous for early cut-off and high grades of expansion —say cut-off below $\frac{1}{4}$ stroke.

THE INDICATOR.

Q. Does a correct diagram curve necessarily show an economical engine?

A. A correct curve is not proof positive of an economical engine, since leakage out may balance leakage in, and not affect the diagram.

Q. Does an incorrect or "bad" indicator diagram necessarily denote a wasteful engine?

A. An incorrect indicator diagram (if correctly taken) of necessity denotes that the engine is *at least* as wasteful as the diagram shows it to be, and perhaps more so.

Q. What are some of the uses of the indicator?

A. The indicator shows the performance, condition, power and economy of the steam engine; the power wasted by want of lubrication, improper alignment of shafting, badly designed gearing, slip, or excessive tightening of belts. It can be used to register the amount of power consumed by each tenant or machine; detects carelessness or incapacity of the engine runners points out leaks, chokes, bad packing, condensation, uneven or badly-timed valve motion, etc.

Q. Are there any other means of measuring power?

A. "The Prony Brake," or friction brake, and the belt or pulley dynamometer measure with sufficient accuracy the power developed by a motor, are invaluable to check the indicator, and measure the power of water-wheels and windmills, to which the indicator cannot be applied.

(Cases are frequent where a turbine, guar-

anteed to develop 85 per cent. of a water-power, is proved by the brake or the dynamometer to give less than 50 per cent.)

Q. Is there any commercial use for the indicator?

A. Engines rated by their builders at 100 horse-power with 22½ lbs. of steam per hour per horse-power are sometimes found by indicator or brake to develop but 75 horse-power, consuming 30 pounds of steam hourly. In each case 2,250 pounds of steam is used, and probably 250 pounds of coal burned; but in the second instance both power and economy are too low, and might sometimes be brought to proper capacity and duty simply by resetting the valves.

A saving of 10 per cent. in cost of lubricants has been shown by the indicator to cause an increase of 10 per cent. in the more important item of coal.

Power lessees paying for 50 horse-power sometimes get but 30; and others, while paying for only 30, use 50. Lawsuits from these causes are frequent, and bad feeling, annoyance and pecuniary loss much more so.

Q. What economy is there in heating the feed water?

A. The following table is given as showing the saving in heating to different temperatures from various temperatures.

[Sometimes these theoretically calculated savings are not reached; sometimes they are somewhat exceeded; the latter case, where heating the feed water causes deposit of scale or of mud in the feed-heater instead of in the boiler.]

TABLE SHOWING SAVING BY HEATING THE FEED WATER.

FINAL TEMPERATURE.	INITIAL TEMPERATURE OF THE FEED WATER.					
	32°	40°	50°	60°	70°	80°
60°	2.39	1.71	0.86	.. ..	.. ..	.. ..
80°	4.00	3.43	2.59	1.75	0.88	.. ..
100°	5.79	5.14	4.32	3.49	2.64	1.78
120°	7.50	6.85	6.05	5.23	4.40	3.55
140°	9.20	8.57	7.77	6.97	6.15	5.32
160°	10.90	10.28	9.50	8.72	7.01	7.09
180°	12.60	12.00	11.23	10.46	9.68	8.87
200°	14.30	13.71	13.00	12.20	11.43	10.65
220°	16.00	15.42	14.70	14.00	13.19	12.33
240°	17.79	17.13	16.42	15.69	14.96	14.20

TABLE SHOWING SAVING BY HEATING THE FEED WATER.—*Continued.*

FINAL TEMPERATURE.	INITIAL TEMPERATURE OF THE FEED WATER.						
	90°	100°	120°	140°	160°	180°	200°
60°	.. ..	.. ..	.. ..	.. ..	.. ..	.. ..	....
80°	.. ..	.. ..	.. ..	.. ..	.. ..	.. ..	....
100°	0.90	.. ..	.. ..	.. ..	.. ..	.. ..	....
120°	2.68	1.80	.. ..	.. ..	.. ..	.. ..	....
140°	4.49	3.61	1.84	.. ..	.. ..	.. ..	....
160°	6.26	5.42	3.67	1.87	.. ..	.. ..	....
180°	8.06	7.23	5.52	3.75	1.91	.. ..	....
200°	9.85	9.03	7.36	5.62	3.82	1.96	....
220°	11.64	10.84	9.20	7.50	5.73	3.93	1.98
240°	13.43	12.65	11.05	9.37	7.64	5.90	3.97

HORSE-POWER.

Q. What are the elements absolutely required in order to calculate the horse-power of an engine?

A. In order to calculate the horse-power of an engine we must know the following elements:

Mean effective pressure.
Length of stroke.
Area of piston.
Rotation speed.

Q. How much deduction from the calculated horse-power of an engine should be made for friction, cylinder condensation, leakage, &c.?

A. The deduction for friction, leakage, cylinder condensation, &c., varies. The larger and better the engine and the better its condition, the less the allowance. As a rough figure, say from 20 per cent. down to 12½ per cent.; or ⅕ to ⅛.

The design of the engine greatly influences the amount of friction. In the supplement to this work [now (January, 1886) in type and to be shortly issued, uniform in size, style and price with this volume, by the same publishers] the marked influence of the position of the fly-wheel and of the engine design (whether vertical or horizontal, etc.) is shown, from actual experiments on a practical scale.

Q. How large an engine will give, say, 18 horse-power?

A. As the horse-power of an engine depends upon so many elements, this question cannot be answered in a general way. The following list shows a number of engines and conditions that will yield, in theory 18 horse:

Cyl. diam., inches.	Stroke, inches.	Rev. per min.	Clearance propor'n.	Cut off.	Actual exp. rate.	Initial pres. ab. atm.	Initial pres. ab. vacuum.	Average total pres.	Back pres. ab. vac.	Mean eff. pres.	Horse-power.
6	12	300	.02	.25	3.78	55.3	70.	42.63	7.7	34.93	17.96
8	12	169	.02	.25	3.78	55.3	70.	42.63	7.7	34.93	17.99
8	12	295	.02	.25	3.78	30.78	45.48	27.7	7.7	20.00	17.97
6	12	212	.02	.25	3.78	80.	94.7	57.67	7.7	49.97	18.
7	12	97	.02	.25	3.78	80.	94.7	87.19	7.7	79.49	17.98
7	12	150	.02	.25	3.78	50.	64.7	59.57	7.7	51.87	18.15
8	12	115	.02	.25	3.78	50.	64.7	59.57	7.7	51.87	18.17
6	18	200	.05	.625	1.56	40.4	65.1	60.22	7.7	52.52	18.
6	18	200	.02	.25	3.78	84.18	98.88	60.22	7.7	52.52	18.
9	9	190	.05	.625	1.56	29.03	43.73	40.76	7.7	32.76	18.
9	9	190	.02	.25	3.78	51.72	66.42	40.76	7.7	32.76	18.
10	10	180	.05	.625	1.56	20.87	35.57	32.91	7.7	25.21	18.
10	10	180	.02	.25	3.78	39.34	54.04	32.91	7.7	25.21	18.
9	9	190	.05	.625	1.56	39.84	54.54	50.46	17.7	32.76	18.
9	9	190	.02	.25	3.78	68.16	82.86	50.46	17.7	32.76	18.
9	9	190	.05	.625	1.56	50.65	65.35	60.46	27.7	32.76	18.
9	9	190	.02	.25	3.78	84.58	99.28	60.46	27.7	32.76	18.
10	10	180	.05	.625	1.56	31.68	46.38	42.91	17.7	25.21	18.
10	10	180	.02	.25	3.78	55.76	70.46	42.91	17.7	25.21	18.
10	10	180	.05	.625	1.56	42.49	57.19	52.9	27.7	25.2	18.
10	10	180	.02	.25	3.78	62.18	86.88	52.9	27.7	25.2	18.
9	15	140	.05	.625	1.56	22.46	37.16	34.28	7.7	26.67	18.
9	15	140	.05	.625	1.56	33.27	47.97	44.38	17.7	26.67	18.
9	15	140	.05	.625	1.56	44.07	58.77	54.38	27.7	26 67	18.
9	14	150	.05	.625	1.56	25.46	37.16	34.38	7.7	26.67	18.
9	14	150	.05	.625	1.56	33.27	47.97	44.38	17.7	26.67	18.
9	14	150	.05	.625	1.56	44.07	58.77	54.38	27.7	26.67	18.
6	14	200	.05	.625	1.56	52.38	67.78	62.72	17.7	28.27	18.
8	8	250	.05	.625	1.56	31.94	46.64	43.15	7.7	50.27	18.
8	8	250	.05	.625	1.56	42.75	57.45	53.15	17.7	50.27	18.
8	8	250	.05	.625	1.56	53.56	68.56	63.15	27.7	50.27	18.
9	9	150	.05	.625	1.56	38.48	53.18	49.2	7.7	63.62	18.
9	9	150	.05	.625	1.56	48.28	63.98	59.2	17.7	63.62	18.
9	9	150	.05	.625	1.56	60.09	74.79	69.2	27.7	63.62	18.

Q. In usual computations, how are these four elements generally expressed?

A. Mean effective pressure is generally expressed in pounds per square inch, length of stroke in feet, piston area in square inches, and rotation speed in turns per minute.

Q. What is the "factor of horse-power?"

A. "Factor of horse-power" is a conventional term used in calculation, and means the product of mean effective pressure, and area and speed of the steam-piston, divided by 33,000. Thus when the area of piston is expressed in square inches, and its speed in feet per minute, the so-called "factor of horse-power," multiplied by the mean effective pressure in pounds per square inch, gives the horse-power of the engine.

Q. What is a horse power?

A. An English horse-power, such as we reckon by,* is the power required to raise 33,000 lbs. one foot high in a minute, or 550 lbs. one foot high in a second, or 1,980,000 lbs. one foot high in an hour.

* A French horse-power is 75 kilogrammeters per second. As a kilogrammeter is 7.233 foot pounds, the French "cheval-vapeur" is only 542.5 foot pounds per second.

Q. How much work will the evaporation of one cubic inch of water, at a pressure of 14.7 lbs. per square inch, do?

A. One cubic inch of water evaporating at 14.7 lbs. per square inch, and making 1,641.5 cubic inches of steam at 212° F. = 100° C., would, if in a vertical cylinder of one square inch bore, raise the 14.7 lbs. 1,641.5 — 1 = 1,640.5 inches = 136.7 feet, doing $14.7 \times 136.7 = 2{,}009.49$ foot pounds of work (not allowing for friction nor for weight of piston).

Q. How many horse-power ought we to get in, then, by the perfect combustion of one pound of coal in an hour?

A. Supposing one lb. of coal to give out in its complete and perfect combustion, to carbonic acid, 14,000 heat units, this is the equivalent of $14{,}000 \times 772 = 11{,}008{,}000$ foot pounds; and this equals $\frac{11{,}008{,}000}{33{,}000 \times 60} = 5.56$ horse-power if exerted in one hour. If exerted in a minute, it would yield

$$\frac{11{,}008{,}000}{33{,}000} = 333.6 \text{ horse-power.}$$

THE GOVERNOR.

Q. What is a governor?

A. A governor is a device by which the speed of the engine is automatically regulated and kept constant, by the rotation of the engine itself.

Q. What are the principal types of governor?

A. The principal types of governor are the "throttler" which keeps the speed uniform by varying the initial *pressure*, and the "cut-off" which varies the *volume* of steam by altering the point of cut-off.

Q. Which is the more efficient, the "throttling" or the "cut-off" governor?

A. The cut-off governor is much the more efficient, but can be applied only to engines so arranged that the point of cut-off may be instantly changed while running.

Q. What are the advantages of the ordinary slide valve engine with throttling governor?

A. The advantages of the throttling slide valve engine, are low first cost, and simplicity.

Q. What is the advantage of having the

governor on the main shaft driving the cut-off mechanism?

A. The advantage gained by having the governor on the main shaft, driving the cut-off mechanism positively, is that there is no danger of the engine becoming detached from the governor, and there can be no irregularity of motion, nor too slow governor speed, by reason of belt slippage, etc.

Q. What is the objection to having the governor on the main shaft, directly actuating the cut-off mechanism?

A. That the governor has to do the actual work of operating the cut-off, instead of simply indicating when the cut-off is to take place.

Q What is a prime requisite of a governor?

A. It is essential that the governor shall be readily adjustable for various desired speeds of engine.

Q. What is a "stop motion" attachment to a governor?

A. A "stop motion" is a device by which should the governor belt break, or the load be too suddenly or too greatly decreased for the governor to handle the engine the

steam will be completely shut off, and the engine brought to a stand still.

Every governor should have a stop motion attachment, or it is incomplete and inefficient.

Q. Of what use is the centrifugal governor?

A. The centrifugal governor saves steam and keeps the rotation speed nearly regular.

Q. What radical defect in the centrifugal governor?

A. The centrifugal governor has this radical defect that the engine "must go fast in order to go slow," that is, speed and not load is the governing element.

Q. What is the undesirable effect of governing by speed instead of by load?

A. In governing by speed instead of by load, the engine does not "regulate regularly," but on commencing to slow, first slows too much, then runs too fast, then too slow again, etc., vibrating between too fast and too slow until the proper mean speed is gradually worked to.

Q. Does the fact that an engine will run a given number of turns—say 60—in a

minute, whenever counted, prove that the rotation speed is regular?

A. The fact that an engine makes a given number of turns—say 60—in a minute, whenever counted, is no indication that the speed is regular. In the first quarter of a minute it may make 16 turns, in the next, 15; in the next, 14; and in the next, 15 again; vibrating in speed in this way all the time.

Q. What is the correct system of testing rotation speed?

A. The correct method of testing rotation speed is to show the speed at each revolution, by a device which will show that one revolution is made, say in $\frac{1}{60}$ of a minute, the next in $\frac{1}{68}$ of a minute, and so on.

DESIGN AND CONSTRUCTION.

Q. What is a "four-ported" engine?

A. A four-ported engine admits the steam through one port and exhausts it through another at the same end.

Q. What are the objections to a four-ported engine?

A. A four-ported engine if it leak after cut-off will blow clear through into the exhaust. In some four-ported engines the

clearance is greater than in some two-ported ones.

Q. Should the steam and the exhaust valves receive their motion together or independently?

A. It is well where there are separate valves for admission and exhaust, that these should receive their motion independently of each other.

Q. What is a good precaution with regard to the throttle?

A. It is well to have two throttles, one of which will control the steam supply should the other stick open or leak.

Q. How is the least possible friction of valves of Corliss type engines secured?

A. To get the least possible valve friction of Corliss type engines, the valves should not bear on their seats, but be hung on trunnions so as to clear them and obviate the gouging action.

Q. In which direction should the steam press on a throttle valve?

A. The steam should be on the side to press the throttle valve to its seat.

Q. What may be said about balanced slide valves?

A. Balanced slide valves, while lessen-

ing the waste of power required to move the valve, are apt to wear out and leak after a few months' use, and become inoperative.

Q. What are the advantages of horizontal engines over vertical?

A. Horizontal engines are generally more accessible than vertical.

Q. What are the disadvantages of the beam engine?

A. The objections to the beam engine are its complication, size, weight and inapplicability to the horizontal type.

Q. What are the advantages of the beam engine?

A. The beam engine has the merit for large powers, of being better balanced than a horizontal engine; and of not wearing out of round in the bore.

Q. What are the advantages of vertical over horizontal engines?

A. Vertical engines take up less room, require less foundation, and wear less out of round in the bore, than with horizontal engines.

Q. What are the advantages of "direct connected" engines?

A. The advantages of direct connected engines are that the first cost and mainte-

nance of belts, pulleys, gears and shafts, the noise of gear and the slip of the belts are lessened, and less space is taken up.

Q. What is the principal mechanical advantage of short stroke?

A. The principal mechanical advantage of short stroke is stiffness.

Q. What are the advantages of long stroke?

A. The merits of long stroke are low speed and friction of the journals and crank pin with the same weight of fly-wheel; but as long stroke engines should have proportionately heavy fly-wheels,* this brings in, where the fly-wheel is heavy enough, increased journal friction due to weight. The longer the stroke the less the *proportion* of clearance volume.

Q. In building slow speed engines, what is aimed at?

A. In building slow speed engines, the effort is made to get light reciprocating parts and heavy fly-wheel rim.

Q. In building high speed engines, what is aimed at?

A. In building high speed early cut-off

* Note under "Proportions."

engines, the effort is made to get heavy reciprocating parts so as to get smooth running.

Q. What is the proper place for the cross head pin, and why?

A. The cross-head pin should be in the centre of length of the shoes, instead of towards one end as is usual, in which latter case there is a tendency to break, and to wreck the cylinder.

Q. What adjustment should the slides have?

A. The slides should have adjustment for wear.

Q. How should the eccentric be fastened to the shaft?

A. The eccentric should be fastened by a key and not by a set screw, as the latter are liable to slip or work loose, and cause much trouble and perhaps great damage.

Q. Should the connecting rod brasses be bored an exact fit to the crank pin and cross-head pin?

A. The connecting rod brasses may well be bored a trifle large for the pin, so that when brought up snug they may leave loose spaces which will hold and distribute the lubricant. A better plan is to make the

brasses an exact fit to the pins, and then file away the bearing on each side at the joint a distance equal to one-twelfth of the circumference.

Q. Which are the best, stud bolts or through bolts for cylinder heads and valve chest covers?

A. Through bolts are to be preferred in every case where they can be used.

Q. What conditions tend to cause leaky valves?

A. Leakage of valves is liable to occur in time with cylindrical valves; with those having variable travel, and where the valve-closing action is not positive. Double poppet valves may leak from unequal expansion.

Q. How should a slide valve be fitted to its seat?

A. A slide valve should be fitted to its seat by filing and scraping them, and *not* by grinding them with emery and oil.

Q. How should the main journal and its bearing be fitted and kept to a bearing?

A. By filing and scraping; when necessary, by chipping, but *never* by using emery.

Q. To insure perfect accuracy in size,

shape and finish of the journals and crank pin, how should they be got up?

A. To make sure that the crank pin, journals, etc., are round, straight and perfectly finished, they should be ground on centres.

Q. What is the best kind of packing for the piston rod and valve rod?

A. The piston rod and valve rod, if small, may be packed with braids of hemp or of cotton wicking; if large, with rings cut from "coil packing" of hemp, asbestos, etc., and if very large, with rings of anti-friction metal, coned, split, and set "breaking joint."

Q. What is the best material for piston packing?

A. If the piston is spring-packed or self-packed, either gun metal or cast iron rings properly split and "breaking joint" will do.

Q. What is the proportion between maximum and average piston speed?

A. Maximum piston speed is 1.5708 times the average. Thus: if an engine has 1,000 feet piston speed in a minute, the fastest speed is at the rate of 1570.8 feet per minute.

Q. Does high rotation speed cramp the open port area proportioned to piston area?

A. High rotation speed does not cause diminished open port area, compared with the piston area.

Q. What is the best method of securing steadiness of rotation speed?

A. The best way to get steady rotation speed is to have a very heavy, well-balanced fly-wheel.

Q. How much work should the fly-wheel store up?

A. Watt made the fly-wheel store up the work of 7½ strokes. Bourne said 6. I should prefer 10 to 12.

Q. Does not a heavy fly-wheel take more power to run it than a light one?

A. If it were not for friction in the main bearings, a heavy fly-wheel would take absolutely no more power to run it than a light one. As it is, the friction of the main journals is increased in the proportion of the increase in fly-wheel weight.

Q. What precaution should be taken in increasing fly-wheel weight?

A. In ordering a heavier fly-wheel, the shaft should be stouter and journals longer.

Q. What precaution should be taken in

starting up and stopping an engine with an especially heavy fly-wheel?

A. When the fly-wheel is especially heavy, the engine should be started or stopped more gradually than with a light fly-wheel.

PROPORTIONS OF PARTS, ETC.

Q. How should a cylinder be bored?

A. A cylinder should be bored vertically, finishing to size with a shallow broad cut and coarse feed, and *invariably without stopping*. Large cylinders should always be bored in a vertical position.

Q. What is the proper thickness of cylinder?

A. Weisbach gives as the proper thickness of cylinder, .00033 times the boiler pressure in pounds per square inch, times the cylinder diameter in inches, plus 0.8 inches. Reuleaux gives the thickness as 0.8 inches plus $\frac{1}{100}$ the cylinder diameter. For a locomotive cylinder 16 inches diameter, 130 pounds boiler pressure, Weisbach's rule would give the thickness as $(.00033 \times 130 \times 16) + .8$ inches, which would be 1.48, or practically $1\frac{1}{2}$ inches.

Q. What shape of cylinder head would

give the greatest strength with a given amount of metal, or require least metal for a given strength?

A. To get greatest strength with a given thickness, or require least metal for a given strength, the cylinder heads should be hemispherical, but this would be too impractical.

Q. How may cylinder heads be strengthened or stiffened?

A. Cylinder heads may be strengthened or stiffened by radial ribs.

Q. What should be the thickness of flat unribbed cylinder heads?

A. Cylinder heads, *if flat*, should have as thickness .003 times the bore, times the square root of the boiler pressure. (Weisbach.) This would give for a 16-inch locomotive cylinder, under 130 pounds boiler pressure, $.003 \times 16 \times 11.4018 = .5473$ inches, which is thinner than the cylinder walls. For a stationary engine 16-inch bore, 81 pounds initial pressure, the Weisbach rule would give the cylinder head thickness as $.003 \times 16 \times 9 = .432$ inches, while the cylinder itself would be $(.00033 \times 81 \times 16) + 0.8$ inches, $.4277 + .8 = 1.2277$ inches, of which the .8 is to allow for reboring, etc.

Q. What causes the greatest strain upon cylinder heads, and why?

A. The greatest strain upon cylinder heads is caused by water in the cylinder; because this is incompressible.

Q. How may danger from water in the cylinder be lessened?

A. Danger from water in the cylinder may be lessened by felting and trapping the steam pipe, lagging or steam-jacketing the cylinder, having the exhaust ports at the side or bottom, having relief-valves, or making frequent use of the cylinder cocks, especially on stopping and starting. Large cylinder heads may have hand-hole plates weaker than the rest of the head.

Q. What should be the thickness of cylinder head bolts?

A. The cylinder head bolts should be in diameter half the width of the cylinder flange.

Q. How far apart should the cylinder head bolts be?

A. The cylinder head bolts should be close enough together not to permit of leakage between head and flange of cylinder. Marks gives as the number of bolts required: .7854 times the square of the

cylinder diameter in inches, times the boiler pressure, divided by 5,000 times the area (in square inches), of a single bolt of the assumed diameter; or, .0001571 times the square of the cylinder diameter, times the boiler pressure, divided by the area of one bolt, a rule which we would prefer to express either as the square of cylinder diameter, times the boiler pressure, divided by 5,000 times the square of the bolt thickness; or, as piston area times boiler pressure, divided by 5,000 times bolt area. This rule would give for a 16-inch cylinder with 130 pounds boiler pressure, and using ¾-inch bolts,

$$\frac{256 \times 130}{5{,}000 \times .5625} = 11.8 \; ;$$

that is, 12 bolts.

Q. Should the steam chest be large or small, and why?

A. Where the steam pipe is large, the chest had better be small, to lessen condensation; but where the pipe wire draws the steam, it is best to have a large chest to prevent falling of pressure in the cylinder, before cut-off.

Q. How thick should the steam chest be?

A. The steam chest should in theory have as thickness one-sixtieth the product of square of inside length and breadth of chest times the square root of the quotient of the boiler pressure, divided by the sum of the fourth powers of the inside length and breadth of chest. (Weisbach.) Thus: a chest 20x16 inches inside, under 80 pounds boiler pressure, should be in thickness

$$\frac{400 \times 16}{60}\sqrt{\frac{80}{160{,}000 + 65{,}536}} = 2.1$$

inches, nearly.

Q. What should be the minimum steam port area?

A. The greater the piston speed, the greater the steam port area required. For a piston speed of 500 feet per minute (corresponding to 250 revolutions per minute in engines of 12-inch stroke), the steam port area should be $\frac{1}{10}$ the piston area. Thus: an engine 9x12 running 250 turns, should have a steam port area of 6.39 square inches, and in the same proportion for higher piston speeds.

Q. What sized ports should be given for high speed locomotives?

A. Engines Nos. 15 and 16, C. & A. R.R.,* have cylinders 17x22 and 66-inch drivers. Hence, to make 60 miles an hour, they would require to make 304 turns per minute = 1,115 feet piston speed. The piston area is 226.98 square inches. The steam port is $1\frac{5}{8} \times 16$ inches = 26 square inches area, and the exhaust port is 3×16 inches = 48 square inches; the valve is of the Allen type, with $\frac{7}{8}$-inch lap and $5\frac{1}{2}$-inch maximum travel. With full port opening, port area is to piston area as 1 to $\frac{227}{26}$, or 1 to 8.73, and would call for a steam velocity in the port of $1{,}115 \times 8.73 = 9{,}734$ feet per minute, supposing full travel and port full open. Engines 17 and 18, C. & A. R.R.,* have cylinders 16x24 inches, and 66-inch drivers. To make 60 miles per hour, they would have to make 304 turns per minute = 1,216 feet piston speed. The piston area is 201.06 square inches; steam ports, $1\frac{1}{4} \times 15$ inches, = 18.75 square inches; exhaust port, $2\frac{1}{2} \times 15$ inches = 37.5 square inches. Proportion of steam port area to piston area being

$$1 \text{ to } \frac{201.}{18.75} = 1 \text{ to } 10.71,$$

* Baldwin build.

the steam would have to travel, supposing ports full open, 1,216 × 10.71 = 13,023 feet per minute. (The valves in these engines are Allen type, $\frac{7}{8}$-inch lap and $4\frac{3}{4}$-inch maximum travel.) The 4-cylinder locomotive, Henry F. Shaw,† has cylinders $10\frac{1}{2}$x24, and drivers 63 inches diameter. To make 60 miles per hour, she would have to make 320 turns (= 1,280 feet piston speed) per minute. Her steam ports are $7\frac{1}{2}$ × 1 inches = 7.5 square inches, and as her cylinder area is 86.59 square inches, steam would have to travel

$$\frac{1,280 \times 86.59}{7.5} = 14,771 \text{ feet per minute,}$$

even supposing full port opening. The port area being to piston area only as 1 to 11.54, the engine is maifestly choked.

For the piston speeds above cited, Nos. 15 and 16 should have

$$\frac{226.98 \times 1,115}{6,000} = 42.18 \text{ square inches};$$

say $16x2\frac{5}{8}$ inches, making the travel $7\frac{1}{2}$ inches.

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† Hinkley build.

Nos. 17 and 18 should have

$$\frac{201.06 \times 1{,}216}{6{,}000} = 40.75 \text{ square inches.}$$

say $15 \times 2\frac{7}{10}$, making travel 7.65 inches; or better, 16×2.55, making travel 7.35 inches.

The "Shaw" should have

$$\frac{86.59 \times 1{,}280}{6{,}000} = 18.47 \text{ square inches,}$$

making $7\frac{1}{2} \times 2.04$ inches; or better ye, 10×1.85, which, with $\frac{3}{4}$ lap, would make the travel required to just uncover the ports, $2 \times (1.85 + .75) = 5.2$ inches, instead of $3\frac{1}{2}$ inches as at present.

Q. Which require the thickest piston heads, vertical or horizontal engines?

A. Horizontal cylinders call for thicker piston heads than vertical.

Q. Which require the thickest piston heads, high or low speeds?

A. High piston speeds call for thick piston heads; but where the piston speeds are equal, the rotation speed has little influence on the necessary piston head thickness.

Q. What should be the piston head thickness?

A. The piston head thickness should be about the fourth root of the product of the piston diameter and stroke. Thus for a 20x48 engine it should be

$$\sqrt[4]{960} = \sqrt{30.98} \quad \text{nearly } 5.57 \text{ inches.}$$

Q. What is the advantage in having a light piston head?

A. The advantage in having a light piston head is that in case of a nut or of a great amount of water getting in the cylinder, the piston head would be smashed rather than the cylinder head knocked out, which would wreck the engine.

Q. What should be the diameter of the piston rod?

A. The diameter of a wrought iron piston rod should be 0.0179 the piston diameter in inches, times the square root of the steam pressure in pounds per square inch. When stroke and diameter are equal, if this comes less than $\frac{1}{12}$ stroke, make it 0.03901 times cylinder diameter times fourth root of steam pressure.

Thus if we have a 20x48 inch with 80 lbs. initial pressure, the piston rod, if of wrought iron, would be equal to

$$20 \times 0.0179 \times \sqrt{80} =$$
$$20 \times 0.0179 \times 8.9443 =$$
$$3.202 \text{ inches.}$$

In a 20x20 inch, it would be $20 \times 0.0179 \times \sqrt{80}$, according to this last rule, which would make it 3.202 inches, which being more than $\frac{1}{12}$ stroke, will do.

In a 12x12 inch, at 80 lbs. boiler pressure this rule would give $12 \times .0179 \times \sqrt{80} = 12 \times .0179 \times 8.9443 = 1.885$ inches.

By the second rule, it would be $0.03901 \times 20 \times \sqrt[4]{80} = 0.03901 \times 20 \times 2.989 = 2.332$.

Q. What should be the material of the piston rod?

A. The piston rod should be of steel, or of hammered wrought iron. "Cold rolled" iron makes good rods, and has greater strength to resist tension and compression than hammered or hot rolled iron.

Q. Speaking approximately, what are the relative diameters of wrought iron and of steel piston rods to do the same work?

A. A steel rod will have 0.9 the diameter and 0.81 the area and weight of a wrought iron one to do the same work, if we consider liability to buckle. (MARKS.)

Q. How should the piston rod be fastened to the head?

A. The rod may be fastened to the piston rod (1) by a key (2) by a nut; and in either case the portion in the head may be either straight, coned towards the cross-head end, or coned towards the back end.

Q. What should be the slide area?

A. The slide area should equal the pressure upon the guides in pounds, divided by 40 to 80.

Q. What should be the distance between guides in engines having a vertical cross-head?

A. Where the cross-head is vertical, the guides must be so far apart that the connecting rod shall clear them at all crank angles; and if the connecting rod cannot be slid out of the guides, then there must be room enough to get the stub end key in and out.

We must add the thickness of the connecting rod to 2.6 times crank radius divided by the number of crank lengths in the connecting rod.

Q. What should be the length of the connecting rod?

A. The length of conuecting rod, if of

wrought iron, should be 4 to 8 times the crank length; that is, 2 to 4 times the stroke.

Thus if the stroke is two feet, the connecting rod should be four to eight feet long.

Q. What should be the thickness of the connecting rod?

A. The thickness of the connecting rod may be 0.0179 cyl. diam. times the square root of boiler pressure, or 12.753 times square root of quotient of indicated horse power by product of stroke in inches and strokes per minute. In cylinders where length equals diameter, if this gives less than $\frac{1}{24}$ con. rod length, then use 0.0195 times cyl. diameter times square root of product of boiler pressure by square of ratio between lengths of connecting rod and crank.

Thus in a 20x48 engine with 80 lbs. boiler pressure, the piston rod may be 0.0179

$\times 20 \times \sqrt{80} = 3.202$.

In an 18×48 in. engine indicating 140 horse power, at 75 revolutions per minute, this would make it

$$12.753 \sqrt{\frac{140}{48 \times 75}} = 12.753 \quad \sqrt{.038888} =$$

2.512341 inches.

Q. What should be the cross section of each leg of the strap?

A. Marks gives for each leg of the wrought iron strap, in square inches, .000078 times the boiler pressure, times the square of cylinder diameter in inches.

For a steel strap he gives $\frac{5}{8}$ the area for wrought.

Q. What should be the crank-pin length?

A. Crank-pin length should increase with co-efficient of friction, with mean steam pressure, with number of revolutions, and with square of diameter of cylinder.

Marks gives the length of crank pin as .0000247 times the co-efficient of friction, times the mean cylinder pressure, times the number of single strokes per minute, times the square of piston diameter; or 12.454 times the co-efficient of friction, times the horse power, divided by the stroke in feet.

The co-efficient of friction is probably from 0.05 to 0.5. The latter in marine engines.

For locomotives, crank making 300 turns per minute, and boiler pressure 150, Marks gives crank-pin length in inches as .013 times the square of piston diameter.

Assuming the co-efficient of friction with constant lubrication as 0.05, a 20x48 engine under 40 lbs. mean effective pressure would require for 60 turns per minute a crank-pin $.0000247 \times .05 \times 40 \times 120 \times 400 = 2.37$ inches long, say $2\frac{3}{8}$ inches.

Q. What should be the diameter of crank-pin?

A. Wrought iron crank-pin diameter may be, in inches, .066 times fourth root of product of boiler pressure, cube of crank-pin length, and square of cylinder diameter.

The 20x48 * at 80 lbs. boiler pressure may have crank pin $.066 \sqrt[4]{80 \times 237^3 \times 20^2} = 1.68$.

The Hartford Engineering Co. makes the crank-pin $\frac{3}{10}$ the bore.

Q. What stresses does the main or crank shaft receive?

A. The crank shaft receives at the begin-

* Mean effective pressure assumed as 40 lbs; co-efficient of friction .05; speed 60 turns per minute.

ning of the stroke a thrust from the piston; between stroke ends a twisting stress, plus the thrust from the piston; and if a fly wheel screw propeller, or paddle wheels be attached their weight tends to bend it.

Q. What should be the diameter of the main shaft journal?

A. The Hartford Engineering Co. makes the diameter of the main shaft bearing half the cylinder bore, and the length equal to the cylinder bore.

Q. What should be the length of main bearings?

A. The main bearings should not be less in length than the cylinder bore for usual good practice.

Q. What should be the minimum diameter of steam pipe?

A. For 600 feet piston speed the steam pipe should have $\frac{1}{10}$ the area of the piston; and for other piston speeds, in proportion.

Q. What should be the minimum diameter of exhaust pipe?

A. An exhaust pipe cannot be too large, and should not be less than $\frac{4}{10}$ the diameter of the cylinder.

CARE AND USE.

Q. Where is the most common place for the piston rod to break?

A. The piston rod most commonly breaks at the cross-head; especially if it has a sharp thread on it at this point, and the cross-head pin is in the end of the length of the cross-head.

Q. How is lead or compression altered?

A. Lead is altered by shifting the eccentric, if it be fastened on by a set screw or by a wedge. Where keyed on, it cannot be altered without changing the lap of the valve, unless an offset key is used.

Q. How may the noise of the exhaust be deadened?

A. The noise of the exhaust may by deadened by an "exhaust head" or by an "exhaust nozzle;" the latter having either a wire spiral coil, through the openings of which the steam is discharged, or a series of perforated beads. Neither of these causes back pressure.

Q. Is there any use of preliminary scientific investigation, more than in so-called "practical tests?"

A. One of my clients—an intelligent

"practical man" (a scorner of science)—undertaking a very bold departure in design and construction of motors, before he consulted me, cost his backers several thousand dollars for experimental machines that would not work. *All* this expense could have been saved had the capitalists consulted any reliable expert as to the feasibility of the plan, which proved to be contrary to the laws of nature.

Q. How may irregularity of motion be prevented?

A. Engines having irregular motion, causing great loss in cotton factories, paper and flour mills, etc., can always be brought to proper performance by intelligent treatment, after using the indicator to reveal the cause. Regularity is especially important to those using the electric light.

Q. What are the principal causes of "knocking" or "pounding?"

A. Knocking and pounding may be caused by lack of alignment in cylinders, main-shaft, or crank-pin, or by lost motion at the crank-pin or cross-head pin, or any of the reciprocating parts.

Knocking may be the result of a misfit

of the piston rod in the cross-head or in the piston head.

It may be caused by wet steam, or by foaming, causing water in the cylinders; or by lost motion in the main bearings.

Q. What are the principal causes of hot main and outboard bearings?

A. Hot main and outboard bearings may be caused by lack of alignment, over-pressure, lack of lubrication, grit in the bearings, too high rotation speed, etc. The longer the bearing the less the liability to heating, so long as everything is kept in line.

Q. What is the advantage of an horizontal engine "throwing over" rather than "throwing under?"

A. Throwing over enables better lubrication of the guides, where the engine has horizontal cross-head, and tends to keep the engine down to the bearings, rather than to pull the caps off the boxes—in other words, the thrust is downward towards the foundation, instead of pulling away from it.

Q. What are the principal causes of hot crank pins?

A. The principal causes of hot crank pins are lack of alignment, improper lubrication,

too tight or too slack brasses, grit in the brasses, and improper material or proportions, and, not infrequently, lack of true cylindrical shape, due to poor material or to too much filing in the lathe, after the finishing cut by the tool.

Q. What is the best lubricant for guides and journals?

A. For general purposes sperm answers best; next, winter-strained lard-oil. For high speed and heavy pressures, add finest air-floated plumbago (graphite, blacklead). For cooling heated journals use flour of sulphur and olive oil. For curing badly scored journals, use lead filings.

Q. What are the evils of tight piston head packing?

A. The evils of tight piston head packing are cutting of the cylinder or of the rings, and waste of power by friction.

Q. What are the evils of loose piston head packing?

A. Loose piston head packing permits the steam to blow through.

Q. How should the crank-pin be oiled?

A. The crank-pin should have an automatic or self-feeding oiler, with adjustable feed.

Q. What is a good arrangement for lubricating the main bearing?

The main bearing may have a tallow cup filled with tallow and graphite,* and also a glass automatic oiler, the former having a large tube to supply the journal should the latter get heated.

Q. "Is it practicable to run an engine without oil or other lubricant in the cylinder?"

A. At low pressures and speeds engines properly proportioned and made can be run without either oil or lubricant, and at the highest pressures and fastest speeds dry black lead (graphite, plumbago) can be used.

Q. What are the effects of using animal oil as a cylinder lubricant?

A. Animal oil in the cylinder is apt to corrode its walls; and also to cause foaming and priming, and incrustation, if the exhaust steam is returned to the boiler by being used to heat the feed water by direct contact. Animal oil is apt to cause gumming up of the piston packing.

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* Plumbago, "blacklead."

Q. What are the effects of using mineral oil as a cylinder lubricant?

A. The use of mineral oil as a cylinder lubricant, has been found, in some cases, where the exhaust was used to heat the feed water, to cause leakage in the boiler, especially in the "steam chimney."

Q. What influence should the steam pressure have upon the cylinder oil chosen?

A. The higher the initial steam pressure, the higher "flash point" required of mineral oils, and the less decomposable animal oils should be. Locomotive service is peculiarly exacting in this respect.

SPEED.

Q. What are the advantages of high piston speed?

A. High piston speed gives little opportunity for condensation from external radiation.

Q. What are the disadvantages of high piston speed?

A. High piston speed brings a heavy shock upon the crank and wrist pins and their brasses at each reversal of the stroke, and makes it difficult to keep the piston and guides lubricated.

Q. What are the disadvantages urged against high rotation speed?

A. High rotation speed makes it difficult to properly balance the reciprocating parts.

Q. What is the practical limit to piston speed?

A. The present practical limit to piston speed is placed by Charles T. Porter at 700 feet per minute for 12 inch stroke, and 1,000 feet for 48 inches.

Q. What is the practical limit to fly wheel speed?

A. The practical limit to fly wheel speed is stated by Porter as 2,300 turns for a 3 foot wheel and 700 turns for a 10 foot wheel; these corresponding to say 22,000 feet rim speed.

Q. Why cannot long stroke engines be run at very high speed?

A. Long stroke engines cannot be run at very high speed because the momentum of the reciprocating parts varies directly as their weight and the stroke, and where the stroke is the longest the connecting rod is the heaviest.

Q. What are the usual average piston speeds?

A. Piston speeds are increasing from

year to year. We may say from 400 to 1,000 feet per minute are usual; the most common being 600; corresponding to 300 turns per minute with 1 foot stroke, 200 with 18 inch stroke, and 150 with 2 foot strokes.

Q. What are the disadvantages of low rotative speeds?

A. The disadvantages of low rotative speeds are excessive cylinder condensation, and the necessity of having a larger engine for a given work, at a given pressure, than with rapid rotation. Besides this, most machinery requires much higher speeds than do steam engines, and here the expense for change of rotation speed is less with high engine rotation speed.

Q. What is the tendency as regards piston speed?

A. The tendency seems to be to shorten the stroke and yet to increase the piston speed by increasing the rotation speed.

Q. What can be said about the speed of engines having "releasing gear" valve motion?

A. The rotation speed of "release gear" engines is necessarily slower than that of "positive gear" engines; hence their stroke

must be necessarily longer, in order to get the same piston speed.

MISCELLANEOUS, TABLES, ETC.

Q. What may be said about foundations?

A. Foundations should be large and deep, especially deep; and the larger the pieces or blocks of which they are made the better.

Q. Where should the engine be put?

A. The engine should be put in a light room near to the boilers, but separated from them, so that it shall not be cut by ashes or coal dust.

Q. When, and by whom was the drop cut-off invented?

A. The drop cut-off was invented in 1841, by Sickels.

Q. When was the Corliss valve gear invented?

A. The Corliss valve gear was invented in 1849.

Q. How is the friction of an engine measured?

A. The friction of an engine is measured by throwing off the main driving belt and letting it run at its usual speed, but without any load. Indicator diagrams are then

taken from both ends of the cylinder, and their average mean effective pressure shows the pressure per square inch upon the piston required to move the engine at its regular speed without load. This will vary but slightly from the friction due to the engine itself, with load on, at the same speed, but is not correct for any other speed. It is best to get several diagrams from each end of the cylinder after the engine has got warmed up, and average an equal number from each end.

Q. What are the highest steam pressures of modern practice?

A. The highest pressures are those carried by Perkins: 400 lbs. (27 atmospheres), habitually, and 1,000 lbs. (70 atmospheres), experimentally.

Q. What is the economical limit to steam pressure?

A. Emery places the economical limit of steam pressure at 100 lbs. in practice. Stevens places the theoretical limit at 250 lbs.

Q. What relation between the initial pressure and the point of cut-off?

A. "The point of cut-off for maximum

efficiency lies nearer the beginning of the stroke as steam pressure rises."

(R. H. THURSTON.)

Q. What is the best proportion of stroke to cylinder diameter?

A. There are probably more engines built "2 to 1" than any other way, because building them "square" or having diameter equal to stroke, necessitates, where both their quantities are small, having excessively high rotation speed to get economical or satisfactory piston speed. Thus, a 6x6 inch has to run 600 turns in order to get 600 feet piston speed.

Q. What is a heat unit?

A. An English heat unit is the amount of heat required to raise the temperature of 1 pound of water at or near 39.1° F., one degree. Its mechanical equivalent is 772 foot pounds—one degree Fahrenheit. A French heat unit, or *calorie*, is the amount of heat required to raise one kilogram (2.2046 lbs. av.), one degree centigrade. One calorie equals 3.967 of an English heat unit.

The following tables answer to reduce English heat units to calories, and *vice versâ*:

FOR CONVERTING ENGLISH (FAHR.) HEAT UNITS INTO CALORIES.

	0.	1.	2.	3.	4.
0.	0.	.252	.504	.756	1.008
10.	2.520	2.272	3.024	3.276	3.528
20.	5.040	5.282	5.544	5.796	6.048
30.	7.560	7.812	8.064	8.316	8.568
40.	10.080	10.332	10.584	10.836	11.088
50.	12.600	12.852	13.104	13.356	13.608
60.	15.120	15.372	15.624	15.876	16.128
70.	17.640	17.892	18.144	18.396	18.648
80.	20.160	20.412	20 664	20.916	21.168
90.	22.680	22.932	23.184	23.436	23.688
	5.	**6.**	**7.**	**8.**	**9.**
0.	1.260	1.512	1.764	2.016	2.268
10.	3.780	4.032	4.284	4.536	4.788
20.	6.300	6.552	6.804	7.056	7.308
30.	8.820	9.072	9.324	9.576	9.828
40.	11.340	11.592	11.844	12.096	12.348
50.	13.860	14.112	14.364	14.616	14.868
60.	16.380	16.632	16.884	17.136	17.388
70.	18.900	19.152	19.404	19.656	19.908
80.	21.420	21.672	21.924	22.176	22.428
90.	23.940	24.192	24.444	24.696	24.948

Multiplier, 0.252.

Divisor, 3.968.

FOR CONVERTING CALORIES INTO ENGLISH (FAHR.) HEAT UNITS.

	0.	1.	2.	3.	4.
0.	0.	3.968	7.936	11.904	15.872
10.	39.68	43.64	47.61	51.58	55.55
20.	79.36	83.32	87.29	91.26	95.23
30.	119.04	123.00	126.97	130.94	134.96
40.	158.72	162.68	166.65	170.62	174.59
50.	198.40	202.36	206.33	210.30	214.27
60.	238.08	242.04	246.01	249.98	253.95
70.	277.76	281.72	285.69	289.66	293.63
80.	317.74	321.40	325.37	329.54	333.31
90.	357.12	361.08	365.05	369.02	372.99

	5.	6.	7.	8.	9.
0.	19.84	23.808	27.776	31.774	35.712
10.	59.52	63.48	67.45	71.42	75.39
20.	99.20	103.16	107.13	111.10	115.07
30.	1 8.88	142.84	146.81	150.78	154.75
40.	178.56	182.52	186.49	190.46	194.43
50.	218.24	222.20	226.17	230.14	234.11
60.	257.92	261.88	265.85	269.82	273.79
70.	297.60	301.56	305.53	309.50	313.47
80.	337.28	341.24	345.31	349.18	353.15
90.	376.96	380.92	384.89	388.86	392.83

Multiplier, 3.968.

Divisor, 0.252.

The following tables of Circumferences and Areas of Circles, and of Hyperbolic Logarithms, will be found of daily use:

CIRCUMFERENCE OF CIRCLES.

DIAM.	CIR.	DIAM.	CIR.	DIAM.	CIR.	DIAM.	CIR.
⅛	.3926	10	31.41	30	94.24	65	204.2
¼	.7854	½	32.98	31	97.38	66	207.3
⅜	1.178	11	34.55	32	100.5	67	210.4
½	1.570	½	36.12	33	103.6	68	213.6
⅝	1.963	12	37.69	34	106.8	69	216.7
¾	2.356	½	39.27	35	109.9	70	219.9
⅞	2.748	13	40.84	36	113.0	71	223.0
1	3.141	½	42.41	37	116.2	72	226.1
⅛	3.534	14	43.98	38	119.3	73	229.3
¼	3.927	½	45.55	39	122.5	74	232.4
⅜	4.319	15	47.12	40	125.6	75	235.6
½	4.712	½	48.69	41	128.8	76	238.7
⅝	5.105	16	50.26	42	131.9	77	241.9
¾	5.497	½	51.83	43	135.0	78	245.0
⅞	5.890	17	53.40	44	138.2	79	248.1
2	6.283	½	54.97	45	141.3	80	251.3
¼	7.068	18	56.54	46	144.5	81	254.4
½	7.854	½	58.11	47	147.6	82	257.6
¾	8.639	19	59.69	48	150.7	83	260.7
3	9.424	½	61.26	49	153.9	84	263.8
¼	10.21	20	62.83	50	157.0	85	267.0
½	10.99	½	64.40	51	160.2	86	270.1
¾	11.78	21	65.97	52	163.3	87	273.3
4	12.56	½	67.54	53	166.5	88	276.4
½	14.13	22	69.11	54	169.6	89	279.6

Circumference of Circles.—Continued.

DIAM.	CIR.	DIAM.	CIR.	DIAM.	CIR.	DIAM.	CIR.
5	15.70	½	70.68	55	172.7	90	2.882
½	17.27	23	72.25	56	175.9	91	285.7
6	18.84	½	73.82	57	179.0	92	289.0
½	20.42	24	75.39	58	182.2	93	292.1
7	21.99	½	76.96	59	185.3	94	295.3
½	23.56	25	78.54	60	188.4	95	298.4
8	25.13	26	81.68	61	191.6	96	301.5
½	26.70	27	84.82	62	194.7	97	304.7
9	28.27	28	87.96	63	197.9	98	307.8
½	29.84	29	91.10	64	201.0	99	311.0

AREA OF CIRCLES.

DIAM.	AREA.	DIAM.	AREA.	DIAM.	AREA.	DIAM.	AREA.
⅛	0.0123	10	78.54	30	706.86	65	3318.3
¼	0.0491	½	86.59	31	754.76	66	3421.2
⅜	0.1104	11	95.03	32	804.24	67	3525.6
½	0.1963	½	103.86	33	855.30	68	3631.6
⅝	0.3067	12	113.09	34	907.92	69	3739.2
¾	0.4417	½	122.71	35	962.11	70	3848.4
⅞	0.6013	13	132.73	36	1017.8	71	3959.2
1	0.7854	½	143.13	37	1075.2	72	4071.5
⅛	0.9940	14	153.93	38	1134.1	73	4185.3
¼	1.227	½	165.13	39	1194.5	74	4300.8
⅜	1.484	15	176.71	40	1256.6	75	4417.8
½	1.767	½	188.69	41	1320.2	76	4536.4

Area of Circles.—Continued.

DIAM.	AREA.	DIAM.	AREA.	DIAM.	AREA.	DIAM.	AREA.
⅝	2.073	16	201.06	42	1385.4	77	4656.6
¾	2.405	½	213.82	43	1452.2	78	4778.3
⅞	2.761	17	226.98	44	1520.5	79	4901.6
2	3.141	½	240.52	45	1590.4	80	5026.5
¼	3.976	18	254.46	45	1661.9	81	5153.0
½	4.908	½	268.80	47	1734.9	82	5281.0
¾	5.939	19	283.52	48	1809.5	83	5410.6
3	7.068	½	298 64	49	1885.7	84	5541.7
¼	8.295	20	314.16	50	1963.5	85	5674.5
½	9.621	½	330.06	51	2042.8	86	5808.8
¾	11.044	21	346.36	52	2123.7	87	5944.6
4	12.566	½	363.05	53	2206.1	88	6082.1
½	15.904	22	380.13	54	2290.2	89	6221.1
5	19.635	½	397.60	55	2375.8	90	6361.7
½	23.758	23	415.47	56	2463.0	91	6503.8
6	28.274	½	433.73	57	2551.7	92	6647.6
½	33.183	24	452.39	58	2642.0	93	6792.9
7	38.484	½	471.43	59	2733.9	94	6939.7
½	44.178	25	490.87	60	2827.4	95	7088.2
8	50.265	26	530.93	61	2922.4	96	7238.2
½	56.745	27	572.55	62	3019.0	97	7389.8
9	63.617	28	615.75	63	3117.2	98	7542.9
½	70.882	29	660.52	64	3216.9	99	7697.7

HYPERBOLIC LOGARITHMS.

The common logarithm multiplied by 2.30258509 gives the hyperbolic logarithm, and the hyperbolic logarithm multiplied by 0.43429448 gives the common logarithm.

The following table contains the hyperbolic logarithms for numbers up to 39, which is considered sufficient for application to expansion of steam:

HYPERBOLIC LOGARITHMS.

No.	Hyp. Logarithms.	No.	Hyp. Logarithms.	No.	Hyp. Logarithms.	No.	Hyp. Logarithms.
1	0.00000	**4**	1.38629	**7**	1.94591	**10**	2.30258
1.1	0.09530	4.1	1.41096	7.1	1.96006	11	2.39589
1.2	0.18213	4.2	1.43505	7.2	1.97406	12	2.48491
1.3	0.26234	4.3	1.45859	7.3	1.98787	13	2.56494
1.4	0.33646	4.4	1.48161	7.4	2.00149	14	2.63906
1.5	0.40505	4.5	1.50408	7.5	2.01490	15	2.70805
1.6	0.46998	4.6	1.52603	7.6	2.02816	16	2.77259
1.7	0.53063	4.7	1.54753	7.7	2.04115	17	2.83321
1.8	0.58776	4.8	1.56859	7.8	2.05415	18	2.89037
1.9	0.64181	4.9	1.58922	7.9	2.06690	19	2.94444
2	0.69315	**5**	1.60944	**8**	2.07944	**20**	2.99573
2.1	0.74190	5.1	1.62922	8.1	2.09190	21	3.04452
2.2	0.78843	5.2	1.64865	8.2	2.10418	22	3.09104
2.3	0.83287	5.3	1.66770	8.3	2.11632	23	3.13549
2.4	0.87544	5.4	1.68633	8.4	2.12830	24	3.17805
2.5	0.91629	5.5	1.70475	8.5	2.14007	25	3.21888
2.6	0.95548	5.6	1.72276	8.6	2.15682	26	3.25810
2.7	0.99328	5.7	1.74046	8.7	2.16338	27	3.29584
2.8	1.02962	5.8	1.75785	8.8	2.17482	28	3.33220
2.9	1.06473	5.9	1.77495	8.9	2.18615	29	3.36730
3	1.09861	**6**	1.79175	**9**	2.19722	**30**	3.40120
3.1	1.13140	6.1	1.80827	9.1	2.20837	31	3.43399
3.2	1.16314	6.2	1.82545	9.2	2.21932	32	3.46574
3.3	1.19394	6.3	1.84055	9.3	2.23014	33	3.49651
3.4	1.22373	6.4	1.85629	9.4	2.24085	34	3.52636
3.5	1.25276	6.5	1.87180	9.5	2.25129	35	3.55535
3.6	1.28090	6.6	1.88658	9.6	2.26191	36	3.58352
3.7	1.30834	6.7	1.90218	9.7	2.27228	37	3.61092
3.8	1.33046	6.8	1.91689	9.8	2.28255	38	3.63759
3.9	1.36099	6.9	1.93149	9.9	2.29171	39	3.66356

Q. What is the advantage of setting the fly-wheel close to the crank?

A. When the fly-wheel is set close to the crank there is less strain on the main shaft and less thumping when the pillow-block is loose.*

Q. What is a good material for connecting rods?

A. A good material for connecting rods is mild, hammered, open-hearth steel.

Q. When a connecting rod has only one solid end, which end should it be?

A. If there is only one solid end that should be the crank-pin end, as the easiest put on. The brasses can then be put on from the front side and held by a cap covering the end of the crank-pin.

Q. How wide should the adjusting wedges be in connecting rods?

A. The adjusting wedges should be as wide as the brasses, to prevent upsetting, mashing or straining the latter.

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* Raffard's experiments described at full length in the Supplement or second part of this work, (which see) show a marked saving in engine friction by having the fly wheel at the crank end.

Q. Which is best for cross-head pins; to have them turn in the cross-head or in the end of the rod?

A. It is best to have the cross-head pin turn in the cross-head, because that gives double length of bearing.

Q. Where the cross-head pin is fast in the cross-head and turns in the rod-end, how should it be made?

A. Where the cross-head pin turns in the end of the rod, it had best be cast in one piece with the cross-head, so it cannot possibly work loose.

Q. What is the best iron for cranks and crank-disks?

A. The best iron for cranks and crank-disks is car-wheel mixture, which is a close, hard charcoal iron.

Q. What is a good material for crank-pin?

A. A good plan is to make crank-pins with a mild steel center and hardened steel wearing surface.

Q. Are tapered crank-pins advisible?

A. Tapering the crank-pin gives increased stiffness and greater wearing surface.

Q. What is the best way to ensure the crank-pin being in perfect line with the main shaft, and at right angles with the crank?

A. The best way is to bore the hole for the pin after the crank has been put on, keyed, and finished up.

Q. What are some of the good points of a governor?

A. A governor should permit the ports to be full open at the beginning of the stroke, and should cut off (without changing the lead) at just such a point as will keep the engine speed constant. It should work under greatly varying load, or greatly varying pressure, or both combined, without permitting over two per cent. of variation in speed, from full load to empty running; and should stop the engine in case it does not work safely. It should not have slipping belts, nor high speed, back-lashing gears, and should have no stuffing box and stem likely to stick.

Q. Does the lead generally remain the same when the cut-off is changed?

A. No; it generally increases as the cut-off is made earlier.

Q. With what class of valve-gear is the increase of lead with earlier cut-off most marked?

A. With the Stephenson link-motion.

Q. Why should not the lead be changed when cut-off is changed?

A. As there is a certain amount of lead required to ensure smooth running at each speed, and as varying speeds are not desirable in stationary engines, the engine should be set, when hot, with the proper amount of lead for the regular speed; and this lead should not be changed while that speed remains constant.

Q. What are the good points of underneath steam-chests?

A. Underneath steam-chests let water of condensation out of the cylinder, without giving it much opportunity to get in.

Q. What produces most of the same result as the under-hung steam chest?

A. Valve and ports on the side.

Q. What benefit comes from having an "overhung" cylinder?

A. The "overhung" cylinder can change in length, by reason of varying temperature,

without springing the frame or throwing working parts out of line.

Q. On which end of the stroke does the piston travel fastest?

A. The piston travels faster on the end of the stroke furthest from the crank.

Q. Which should be the weaker; the piston head or the cylinder head?

A. The piston head should be weaker than the cylinder head, as being cheaper, quicker and easier replaced, and doing less damage when broken by a charge of water or other cause. It is, however, a good plan to provide a comparatively weak "breaking piece" in the cylinder head, so that this will give way sooner than the piston head or cylinder head.

Q. What is the advantage of putting the pillow-block caps on at an angle?

A. When pillow-block caps are put on at an angle the construction is very simple, and the adjustment can be made in line with the wearing parts.

Q. What is the disadvantage of drop cut-off engines with a single eccentric?

A. If the cut-off does not take place dur-

ing the forward part of the valve's motion, the valve must remain undisturbed until the next stroke, cut-off will be missed, and steam follow full stroke. This is bothersome, if the work so varies as to require cut-off at from $\frac{1}{4}$ to $\frac{9}{16}$ stroke.

Q. What are the best materials for cross-head shoes?

A. Depends on the cross-head guides, and the amount of pressure, and the kind of lubrication. Cast iron, bronze, and apple wood, all give satisfaction.

Q. Which should have the greater diameter, the crank-pin or the cross-head pin?

A. The crank-pin should be the greater in diameter. For a six-inch crank-pin there would be needed a cross-head pin about 4.4 inches in diameter.

Q. Are there engines with less than 2 per cent. of clearance?

A. There are some of very large size where the clearance runs lower than this; but very few.

Q. If you build an engine 18"×36", or 18"×42", what should be the diameter of the crank shaft, and what that of the crank-pin?

A. The shaft one-half the cylinder diameter, and the crank-pin one-fourth.

Q. What sets the limits on high speeds?

A. Constructive considerations set the limits on high rotation speed.

Q. What is the principal disadvantage of a four-ported engine?

A. The principal demerit in most four-ported engines is that if the steam-valves leak between cut-off and cushion, the live steam will blow through with the exhaust.

Q. Is there more friction for a heavy fly-wheel than for a light one?

A. Not much more.

Q. Could an engine be made with the main valve like that is the Wheelock, but having the cut-off inside of the main valve instead of under it?

A. Engines are so made.

Q. Which is the thicker, the cylinder flanges or the cylinder heads; and why?

A. The cylinder flanges are the thickest, so that a break would occur in the head rather than in the cylinder well itself.

Q. Why should locomotive cylinders have thicker metal than on stationary or marine engines?

A. Because there must be more metal allowed for re-boring.

Q. Why should there be more frequent re-boring in locomotive than in marine or stationary cylinders?

A. Because the piston speed is generally higher, and because there are often ashes drawn in through the exhaust nozzle into the cylinder, which they cut considerably.

Q. What is the objection to bolts in a cylinder head, for locomotive work?

A. If a bolt breaks, it is necessary to take off the whole cylinder lagging in order to place the new bolt in position. But if a stud breaks, the part left in the cylinder flange can be drilled out without disturbing the cylinder lagging.

Q. How thick should the cylinder head be?

A. The cylinder head may have a thickness equal to the square root of the boiler pressure, times 0.003 the cylinder diameter in inches.

This rule is given on good authority; and, like almost all rules, is good within certain limits.

For an 18″ cylinder, this gives the following thicknesses for various boiler pressures:

Boiler Press.	Sq. Rt. B. P.	C H. Thick.	Boiler Press.	Sq. Rt. B, P.	C. H. Thick.
40	6.3246	.34	105	10.2470	.56
45	6.7082	.36	110	10.4881	.57
50	7.0711	.38	115	10.7238	.58
55	7.4162	.40	120	10.9545	.59
60	7.7460	.42	125	11.1803	.60
65	8.0623	.44	130	11.4018	.62
70	8.3666	.45	135	11.6190	.63
75	8.6603	.47	140	11.8322	.64
80	8.9443	.48	145	12.0416	.65
85	9.2195	.50	150	12.2474	.66
90	9.4868	.51	155	12.4499	.67
95	9.7468	.53	160	12.6491	.68
100	10.	.54	165	12.8452	.69

The above thicknesses are got by multiplying the figures in the second columns by 0.054.

The annexed tables gives the required

cylinder head thickness, according to Marks' rule, for various boiler pressures and cylinder diameters :

B.P. LBS.	10" 0.030	12" 0.036	14" 0.042	16" 0.048	18" 0.054	20" 0.060	$\sqrt{\text{B. P.}}$
40	.19	.23	.27	.30	.34	.38	6.3246
45	.20	.24	.28	.32	.36	.40	6.7082
50	.21	.25	.30	.34	.38	.42	8.0711
55	.22	.27	.31	.36	.40	.44	7.4162
60	.23	.28	.33	.37	.42	.46	7.7460
65	.24	.29	.34	.39	.44	.48	8.0623
70	.25	.30	.35	.40	.45	.50	8.3666
75	.26	.31	.36	.42	.47	.52	8.6603
80	.27	.32	.38	.43	.48	.54	8.9443
85	.28	.33	.39	.44	.50	.55	9.2195
90	.28	.34	.40	.46	.51	.57	8.4868
95	.29	.35	.41	.47	.53	.58	9.7468
100	.30	.36	.42	.48	.54	.60	10.
105	.31	.37	.43	.49	.55	.61	10.2470
110	.31	.38	.44	.50	.57	.63	10.4881
115	.32	.39	.45	.51	.58	.64	10.7238
120	.33	.39	.46	.53	.59	.66	10.9545
125	.34	.40	.47	.54	.60	.67	11.1803
130	.34	.41	.48	.55	.62	.68	11.4018

For convenience in extending the table,

the square roots of the boiler pressures are given in a separate column, and under each cylinder diameter is given, in small figures, the factor by which to multiply the square root of the boiler pressure. (This factor is, of course, simply three thousandths of the cylinder diameter.)

Q. How may the weight of fly-wheel rim be determined?

A. The Buckeye Engine Company divides 6,500,000 by the diameter of the wheel in feet and by the square of the number of revolutions per minute to get the weight of rim per horse power in pounds for automatic engines; and for throttling engines substitutes 5,000,000 for 6,500,000.

Thus an automatic engine making 100 turns per minute would require for a 13-foot wheel 6,500,000 ÷ (13×10,000) = 50 pounds per horse power in the rim: and if it was a 250 horse-power engine the weight required in the rim would be 250×50 = 12,500 lbs.

If it was a throttling engine the Buckeye people would give in the rim 5,000,000 ÷ (13×10,000) = 38.46 pounds per horse-power or 38.46×.250 = 9,615 pounds.

The above rule gives much too light weights for rolling-mill work, and the weight should be increased 100 per cent., if transmission is by belts, and 100 per cent. if by gear-wheels, or if there is "direct drive" by clutches.

If the diameter of the fly-wheels be reduced to ten feet, the rim-weight required for the automatic engine at 100 turns per minute would be $6{,}500{,}000 \div (10 \times 10{,}000) = 65$ pounds per horse-power instead of 50, and for the throttling engine at the same speed, $5{,}000{,}000 \div (10 \times 10{,}000) = 50$ pounds per horse power instead of 38.46.

Q. A fly-wheel designed for a certain engine is, say, 18 feet in diameter, and 10,000 pounds rim weight. If lack of space requires a 14-foot wheel, how should it be proportioned?

A. To make a small wheel equal in efficiency to the large one, its weight should be inversely as the square of the diameter to center of rim thickness as compared with that of the larger wheel. If the diameter of large wheel is 18 feet to center of rim, and that of the small wheel 14 feet, then the

weight of the small wheel should be as $18\times 18=324$ is to $14\times 14=196$ compared with the large one. So the 14-foot wheel should have a rim weighing $10,000\times 324 \div 196=17,041$ pounds.

Q. What is a "stroke" of a steam engine?

A. A to-and-fro motion, during a complete crank revolution, from one dead center position to the same.

Q. What are the ways of measuring the stroke of an engine?

A. Measure the distance from crank pin center to shaft center, and double it; scribe a mark on the crosshead slides and measure between the extreme points on the guides, which this mark attains.

Q. How long is the crank throw, compared with the stroke?

A. One-half.

Q. How can you find the exact dead center of an engine?

A. Find that point at which, marks being made on crosshead and guides, partial revolution of the crank causes the crosshead to move toward mid-stroke.

Q. How can you find the mid-stroke of an engine?

A. Put a scribe mark on the crosshead, and one on the guides to correspond, at each dead center; equally divide the distance between the marks on the guides.

Q. Is the piston in the center of the cylinder length when the crank is at half-throw?

A. No.

Q. What is the use of the counter-bore?

A. To prevent the piston from wearing a shoulder in the cylinder, at the end of its stroke.

Q. How does the counter-bore prevent this?

A. This piston travels beyond the beginning of the counter-bore; hence there is nothing for its edge to wear a shoulder on.

Q. What harm would a shoulder do, even if there was one there?

A. If there was any change in the travel of the piston, by which its stroke was increased, or without the stroke being changed in length, it was moved along the bore, either toward the crank, or from it, the edge of the piston would strike this shoulder, and a smash-up would be likely to occur.

Q. What would cause the stroke of the piston to lengthen?

A. This would hardly be likely to happen. It would necessitate the crank pin getting smaller, and at the same time so eccenrically worn as to bring its center further from the main center than it was oaiginally.

Q. What would cause the stroke to change in place in the bore?

A. Such an arrangement of the adjustment in the connecting rod brasses, or in the fastening of the piston rod, either in the crosshead or in the piston head; or such a change in the main bearing brasses, as would move the piston head toward or from the main bearing.

Q. What would be most likely to cause the stroke to move further from the crosshead end of the cylinder?

A. Lengthening of the distance between centers of the crosshead pin and the crank pin, caused by having the adjustment of the brasses such that the key for the crosshead pin was in front of the pin, and that of the crank pin was behind the latter. Then as the brasses wore and the keys were

driven up, the pins would be driven farther apart.

Q. Having provided a counter-bore to prevent accident by the piston head striking any possible shoulder, is there any more danger of accident taking place by change in place of stroke

A. Yes, particularly if there is not very much head clearance. In engines having very small head clearance, there is some little danger lest the piston head strike the cylinder head, and a good deal more liability to there being an accident by reason of water in the decreased clearance space.

Q. What precautions should be taken in packing a stuffing box with soft packing?

A. Cut the lengths a very little too short, even if that allows a very slight leak at first; then when the packing gets wet and swells it will be tight without causing undue friction.

Q. Should all the rings of soft packing be put in before tamping up, or not?

A. The lengths should be put in one by one, and each one properly sent home before the next one is put in.

Q. What may be said of the use of a tamping stick for soft packing?

A. For small packing, each coil should be sent home with the gland or follower, and before the next one is put in; care being taken not to pack too tight. In the larger sizes of packing, say from 1" up, it is good practice to have the packing from $\frac{1}{16}$ to $\frac{1}{8}$ larger than the space, and in such case it may be necessary to use a tamping stick to get the ring started; but you should then follow up with the gland.

Q. What should be done in case the eccentric got worn flat in two spots?

A. Turn or file it down to a true circle.

Q. How would this affect the valve throw?

A. Lessen it.

Q. What is the meaning of "brass bound?"

A. Boxes are brass bound when the half brasses touch each other and can not be driven up any closer.

Q. Suppose the crosshead boxes and crank-pin boxes get brass bound; how may their lost motion be taken up?

A. File off their top and bottom edges.

Q. What should be done when the crosshead shoes get worn?

A. They should be trued up by planing, or by chipping, filing and scraping ; re-babbitted if formerly filled.

Q. What should be done if the cylinder gets worn out of true?

A. It should be re-bored, if thick enough to stand it.

Q. What should be done when a slide-valve gets leaky from wear?

A. It should be re-faced in a planer ; or chipped, filed and scraped true, and its seat either chipped, filed and scraped, or trued by a valve-seat planer.

Q. What should be done when the crank pin or the crosshead pin gets worn out of round?

A. It should be turned true, in place, by a special machine ; or else filed and draw-filed true.

Q. What would you do if a main bearing heated?

A. Slack up the cap bolts, and apply black lead and oil.

Q. What should be done if the eccentric slipped round?

A. It should be put with its "belly," or

full part, 90° ahead of the crank (in the direction in which the crank turns), and then (the crank being put on the center) moved enough further ahead to open the valve the desired amount of lineal lead (as shown by scribe marks, or by opening the chest); or to give the eccentric the desired amount of angular lead. Then the set screws, or the keys, should be fastened.

Q. What should be done if the stop valve stuck open?

A. The back valve at the boiler should be used, if there was one.

Q. Why do stop valves stick on their seats?

A. They sometimes do so because shut tight when cold; then the stem lengthens more than the case, and they are hence jammed hard down.

Q. What should be done if the stop valve stem broke off inside the stuffing box?

A. The back valve should be shut; then the disk of the stop valve removed from its seat, and the engine controlled by the back valve.

Q. Suppose there was no auxiliary stop

valve; how could the engine be stopped or started?

A. In some engines this could be done by disengaging the eccentric hook and working with the starting bar.

Q. What should be done in case a bolt or nut broke off and let steam escape through the hole?

A. Plug it with pine wood; or hold a gum patch over it by an iron washer and a stick jammed against the nearest convenient wall.

Q. What is so called "lost motion?"

A. The space through which one moving piece passes, while moving from contact with its mate, in one extreme position, to contact in the opposite extreme position: familiarly speaking, the amount of "shake" in a joint which should be a good fit.

Q. How may lost motion be taken up?

A. By set screws, wedges, or "shims" (thin pieces of paper, tin, etc.)

Q. Where should knocks be looked for?

A. In connecting rod box, piston, cross-head key, valve connections, pillow block box, crank pin boxes, and crosshead slides;

in rocker arm brasses and connections, where there are any.

Q. When an engine is out of line, where is there most strain, and why?

A. When the piston is furthest from the crank; because then the connecting rod and crosshead connections have least chance to spring.

Q. How should an engine be lined up?

A. Take off both cylinder heads; take out the piston; remove crosshead and connecting rod; make a wooden "spider" to fit each end of the cylinder bore; draw a cord through this, and see that the crank shaft is at right angles to the cylinder axis; and that the crank is also at right angles thereto when on each center.

Q. In resetting an engine, what is one of the first things to do?

A. To clean all parts thoroughly; taking off the back head, and taking out the piston, having driven out the key in the crosshead.

Q. What precaution should be taken with the packing rings in resetting a horizontal engine?

A. The rings should be turned "other side up," so as to bring the "down" side, that has been getting the most wear, to the top of the head, so as to equalize the wear.

Q. How may the packing springs be tested?

A. The packing springs may have their elasticity tested by pressing them with a lever.

Q. Should packing rings be run loose or tight?

A. Packing rings should be run as loosely as will permit of their being steam tight under the maximum steam pressure that will be put on them.

Q. How may the tightness of packing rings be tested?

A. By blocking an engine on the top or bottom quarter and opening the throttle. Then if steam escapes at both cylinder cocks the rings are not tight enough.

Q. In setting packing rings how may the piston rod be kept central?

A. By frequent use of calipers, having one point turned out at right angles to the other; or by a hard-wood distance piece.

Q. How should the valve position be marked?

A. Before putting on the chest cover the valve should be put in its central position and a mark put on its stem, and a corresponding locating mark on some immovable part of the chest — not the stuffing-box gland.

Q. How may the tightness of a slide-valve be tested?

A. The tightness of the slide-valve may be tested by blocking the engine in such a position that the valve will stand centrally over the ports, covering them; (this can be done by means of the scribe marks;) then letting steam into the chest; when, if any steam escapes through the cylinder cocks, the valve leaks.

Q. How may the bearings be "cut" from gum?

A. All gum may be "cut" out from the bearings by the use of coal-oil or turpentine.

Q. How may pounding be searched for?

A. Pounding may be searched for by blocking the cross-head at mid-stroke, and

then working the cross-head slightly back and forth, and watching the bearings.

Q. How may pounding be intensified so as to make it more easily found?

A. By giving the engine its heaviest load.

Q. What scribe marks should be made on the guides?

A. When the connecting rod is down the piston head should be shoved up until it touches each cylinder head, and "danger marks" scribed on the guides, corresponding to some mark on the cross-head, so as to show if the piston head is too near the cylinder heads, when all is keyed up.

Q. How may it be shown if the main shaft is out of line with the cylinder?

A. Lay a level across the guides and mark where the bubble stands; then when laid on the main shaft the bubble should stand at the same point, (supposing that the level has not been turned end for end).

Q. How may the shaft be tested for squareness fore and aft?

A. By stretching a line parallel with the guides, out past the crank; the distance between the line and the crank-pin collar

should be the same on both front and back centers. If there is a crank disk, the distance between that and the line should be the same, fore and aft, no matter where the crank-pin stood.

Q. How should an engine be started; and why?

A. Slowly; first warming up the cylinder; the drip cocks being left open, to let out the water of condensation.

Q. What harm might result from starting with drop cocks shut, or shutting them too soon?

A. The cylinder head or the piston head might be smashed through the inelastic water of condensation filling up all the clearance space.

Q. What precaution is it well to adopt with regard to the main steam pipe?

A. To have a blow-off cock right back of the throttle valve, to drain the pipe of the water of condensation, which might otherwise get into the cylinder and cause trouble or accident.

Q. Should the governor speed of a Harris

Corliss engine be increased or diminished to speed up the engine?

A. It should be decreased.

Q. "We have just got through having our back cylinder head knocked out by the connecting rod having got too long. How may this be prevented in the future?"

A. You should have a gauge to measure the distance between wrist-pin and crank-pin. Have the brassses so that as one end of the rod is lengthened by being set up, the other will be shortened.

Q. Can a leaky piston make much difference in the steam consumption, or is this only a "bug-a-boo?"

A. I have before me the record of a case where I found everything in good condition, except that the piston was leaky, and tested the coal consumption before and after setting out the rings. The coal saving in this instance (the engine was 18″×30″, making 100 turns per minute, and cutting off about half-stroke before the piston was packed,) was the difference between $4\frac{1}{2}$ gross tons for ten hours engine-work, and $2\frac{3}{8}$ tons; or $2\frac{1}{8}$ tons per day, $12\frac{3}{4}$ tons per week, and about

650 tons per year. At $3.50 per ton, that came to $2,275 a year.

Q. Should two or more engines exhaust into the same pipe?

A. When two or more engines exhaust into the same pipe, there is apt to be excessive back pressure in both, or, at least, the one which would naturally have the most back pressure will be apt to increase that in the other. It should be noted in indicating an engine, whether or not its back pressure is likely to be increased, without any fault of its own, by the exhaust from another engine.

Q. Why is it that better results are sometimes got with throttle only partly open than with it full wide open?

A. Perhaps the steam is very wet and the throttle dries it. This question can not be properly answered without fuller particulars being given.

Q. What are the effects of high piston speed on the rod-packing?

A. High speed tends to cause rapid wear of the rod-packing, and leakage of the stuffing-box.

Q. What are the effects of high pressures upon the rod-packings?

A. High pressures increase the difficulty of keeping the valve-stem and piston-rod stuffing-boxes tight.

Q. Is there any good way to prepare piston rod-packing before putting it in the stuffing-box?

A. Take some pieces of wrought-iron pipe somewhat larger in bore than the stuffing-box, and cut packing to fit these; put them, with the packing in them, in the cylinder oil can, for two or three days; take them out and drain them.

Q. If an engine is balanced so that it will run smoothly, will it be safe to put upon the top floor of a building?

A. Not always. The engine may run smoothly at one speed, and at another speed it may shake the whole building, even though that speed is slower than the one at which it runs smoothly.

Q. How is the best general balance of an engine got?

A. When the counter weight equals one-half the weight of reciprocating parts,

increased by that portion of the connecting rod which may be considered as centrifugal in its effect.

Q. Has the steam pressure upon the piston of a steam engine any effect upon the balance of the engine?

A. No.

Q. What is friction?

A. The resistance which two contacting surfaces have to being moved, one over the other.

Q. What are caused by friction?

A. Heating and abrasion, or cutting.

Q. What reduce friction?

A. Lubrication; good fits; proper speed; lessening the pressure between the contacting surfaces.

Q. What is abrasion?

A. Wear, grinding, cutting.

Q. How many kinds of friction are there?

A. Three; sliding, rolling, and fluid.

Q. What kind is the friction of the cross-head on the guides?

A. Sliding.

Q. Then is not that of the journals in their bearings rolling friction?

A. No. The friction of a journal with its bearings is not rolling friction, nor anything like it; it is simply continuous sliding friction.

Q. Where does sliding friction occur?

A. Sliding friction occurs where any one point of either of the rubbing surfaces passes a number of other points upon the other surfaces.

Q. What distinguishes rolling friction?

A. In rolling friction, a number of successive points upon one of the surfaces is presented to a number of successive points on the other. Bearing this in mind, it will readily be seen that where the journal runs in a bearing, or upon it, or under it, (for journals run all three of these ways,) each point upon each of the rubbing surfaces passes every other point upon the circle, of the other surfaces.

Q. Which is the greater, sliding or rolling friction?

A. There is much more resisting force where this continuous sliding friction takes place, than where one wheel runs upon another.

Q. Is friction a very important matter, outside of the question of wear?

A. The importance of reducing journal friction may perhaps be appreciated when we consider that in many machines the only losses of power are those due to its retarding force.

Q. What is the prime cause of friction?

A. It may be set down as a pretty general principle, that the rougher the surfaces in contact, the greater will be the amount of friction. Where the journal rotates in the bearing (or on it, or under it, as the case may be), the roughnesses of one surface interlock with those upon the other.

Q. How may we lessen this sliding friction of journals?

A. We may lessen this friction by lessening the pressure between the surfaces, and thus enabling the roughnesses to ride over one another more easily; or we may fill up the depressions with some substance which will lessen the depth of the projections.

Q. What may be said about the friction of a journal in a bearing having intermittent lubrication?

A. Where a journal runs in a bearing which is lubricated, the friction is not that of solids but partly that of solid upon liquid, and partly that of one liquid film upon another. When the journal runs dry, there is solid friction; when it is flooded there is liquid friction; between these extremes there is mixed friction; and this is generally the result.

Q. Would it be a good thing to have perfect fluid friction under journals?

A. It would be highly desirable to have a condition of perfect fluid friction with journals; for, under such circumstances, the friction should be independent of the pressure, and vary at the square of the speed. It would be directly as the area of the journal and bearing and would be reduced as the diameter rose and the viscosity of the lubricant decreased.

Q. What is the minimum loss from engine friction?

A. That cannot be stated. An engine that has a friction of only 10 per cent. of its developed power is doing very well.

Q. What will help get it down this low?

A. Good regular lubrication.

Q. What is the maximum percentage of friction possible; and where do we find it?

A. 100 per cent.; where the engine is "running empty."

Q. Does friction increase with speed, or decrease?

A. Both. Where the speed is slow, increasing it may decrease the friction per turn; but where the speed is fast, it may be found that increasing speed increases the friction per turn also. There are two sides to almost every question, and particularly so in this case.

Q. What may be said about loss from friction of the slide-valve?

A. In most slide-valve engines there is a great unbalanced pressure upon the valve area, or upon a large portion of it, and this causes friction between the valve and its seat. If there was not a great deal of friction here the slide-valve stem could be made of much less diameter.

Q. What is a good proof that a slide-valve is really balanced?

A. Whenever you hear any one talk

about having a perfectly balanced slide-valve, just ask him to turn down the valve stem for about one inch of its length, anywhere outside of the stuffing box. When you can run an 18" by 36" engine with a quarter-inch valve stem, then you can talk about a balanced valve.

Q. What other places are there where loss is occasioned by friction?

A. There is friction between the piston and the cylinder, especially in horizontal engines. There is friction in the stuffing-boxes. There is friction between the cross-head slides and their guides. There is friction in the main journals; at the crank pin, at the crosshead pin, and all over the engine wherever there are surfaces rubbing together.

Q. Has the diameter of the journal much or little effect upon the friction?

A. Very little.

Q. How about the matter of fit between a journal and its bearings?

A. It is highly desirable that a journal and its bearings should fit each other (although not accurately or tightly) through as large

an arc as possible. Thus, in the case of horizontal journals, it is best that the journal and its under bearing should have an arc of contact of nearly 180°; although it is much more often that we find only 100°.

Q. What are the objections to very loose bearings?

A. It is very important that the bearing should not be much too large for the journal; as in this case the pressure would be concentrated over a smaller area than if the bearing touched the journal all around.

Q. How about a journal "working to its bearing?"

A. It is much better to have the journal made to fit its bearing than to trust to the latter wearing to fit the journal.

Q. Is the pressure on the bearing the same all around its circumference?

A. In a horizontal, semi-circular bearing, the pressure is nothing at the horizontal line, and greatest at the bottom.

Q. As between a too loose journal and a too tight one, which is the greater evil?

A. It has been calculated that where a horizontal bearing and its journals are

absolutely fitted together, the friction is 1.27 times as great as with a loosely fitted journal. Where the journal is so grasped as to give uniform pressure throughout, the friction is 1.57 times as great as it would be if fitted loosely.

Q. What class of engines generally give most trouble with their crank pins?

A. Propeller engines are generally particularly troublesome in this respect, unless the crank pin length is made very great.

Q. What difference does the diameter of a crank pin make, as regards heating?

A. As long as the pressure and speed are reasonable, that is, are not calculated to squeeze out the lubricant and cut the brass, the diameter of the pin does not affect its heating: because while small journals have more pressure per square inch on them, they have proportionately less velocity of rubbing surfaces.

Q. At what point in the stroke does the friction upon the crank pin bear the greatest proportion to the useful effect of the stress in causing the crank pin to turn?

A. At the dead points.

Q. What curious effect has been shown by calculation, with regard to the dead points?

A. That for about three degrees after the crank has left the dead point, the effect of the pressure upon the crank pin is actually to retard the engine.

Q. Are the top and bottom guides of a horizontal engine worn alike?

A. No; in engines which throw over, most of the pressure and wear are upon the bottom guide, and *vice versa.*

Q. How is it that the pressure is always upon one guide, upon both the front and the back stroke?

A. When the engine is throwing over and running forward (speaking of a horizontal stationary engine), upon the out stroke, the resistance of the crank, upon being lifted and forced forward, presses the crosshead against the bottom guide. Upon the return stroke, the resistance of the crank, upon being pulled back and lifted, draws the crosshead against the lower guide.

Q. What may be said about the manner of lubricating?

A. It should be regular, continuous, and uniform

Q. Are "sight feed" cups desirable?

A. Yes; in most places.

Q. Are "automatic" cups advantageous?

A. Yes; in nearly every case.

Q. What may be said about oil holes and passages?

A. Oil holes should be kept closed by wooden plugs, if they have no cups. The passages should be regularly cleaned out so that it may be certain that the oil reaches the surfaces for which it is intended.

Q. What are good coolers for hot bearings?

A. Graphite (which is the same as plumbago, or black lead), flour of sulphur, mercury (quicksilver).

MEMORANDUM.—"An ounce of prevention is worth a pound of cure."

Q. Is isinglass a good lubricant?

A. Mica is doubtless meant. It does for large slides, as ways in shipyards, but not for engine cylinders, and hardly for journals, unless very large ones.

Q. Should engine cylinder oils vaporize with the steam, or not?

A. It is very desirable that cylinder oils should not vaporize with the steam, but should remain unaffected by the high temperature in the cylinder.

Q. What is the effect of the high cylinder temperature on some oils?

A. Oils which vaporize or are otherwise affected by the pressure and temperature generally have their lubricating powers lessened by the change; and especially with those entirely or largely of animal composition, are changed into substances which "pit" or "honey-comb" the surfaces which they should protect from wear or injury.

Q. How should oils be selected for high cylinder temperatures?

A. In a locomotive cylinder, where a pressure of 160 lbs. by the gauge is reached there is a maximum temperature of 371° Fahrenheit; and a mineral cylinder oil to stand this heat must have a "flash point" above that temperature; other things being equal, the higher the better.

Q. Are all oils with high flashing points good cylinder oils?

A. It does not follow that having a high flash point makes a good cylinder oil. There must be "body" to resist pressure; and there must be "lasting properties." Judicious mixtures generally produce oils having the desired qualities in the wished-for proportions, as is shown only by actual test in engine cylinders.

Q. What is the influence of the viscosity or thickness of the lubricant?

A. The more viscous the lubricant, the greater pressure can be carried upon the bearing.

Q. What should be borne in mind in choosing material of journals and of bearings?

A. In choosing the material for journals and bearings, it is best to have one of the rubbing surfaces, at least, of a granular material; and to make that one which is the easier to replace, of the softer material. This one, which is generally the easier to replace, is the bearing.

Q. How can you determine the pressure which a bearing will stand?

A. There is no use figuring upon the pressure which a bearing metal will stand if it exceeds that which the lubricant will bear.

Q. Is there any limit to the power of an engine as long as there is a chance to increase her speed; boiler pressure always the same?

A, Increase of engine speed, without diminishing the mean effective pressure, always increases the capacity in the same proportion.

Q. Why do you so stipulate about the "mean effective" pressure?

A. I say "mean effective" pressure because at considerable increase of piston speed there might be wire-drawing of the live steam, or cramping of the exhaust, or both. You might double the speed of a slow-running engine without cramping her, but when you once reached the limit of speed at which she had port area enough, the "mean effective" pressure would be diminished.

Q. Are there any other limiting conditions in this case?

A. Yes; it is also understood that the speeds are not to be increased so as to cause flapping of the belts, or heating of the wearing surfaces of engine or shafting.

Q. Will any arrangement of of pulleys on engine, countershaft, or machine, increase the power developed by the engine, or decrease the power taken by the machine?

A. No.

Q. If you lighten the class of work on a machine can you speed it up correspondingly without increasing the power it takes?

A. Yes; conversely, a heavier class of work demands more power or slower speed. A faster speed permits increase of onput, but demands more power.

Q. Is there any power in the governor?

A. No, it merely attends to the regulation of the steam pressure, so that under a light load the pressure shall be diminished.

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SUPPLEMENT

TO THE

STEAM ENGINE CATECHISM.

SUPPLEMENT

TO THE

STEAM ENGINE CATECHISM:

A SERIES OF

DIRECT PRACTICAL ANSWERS TO DIRECT PRACTICAL QUESTIONS

MAINLY INTENDED FOR THE USE OF

YOUNG ENGINEERS AND FOR EXAMINATION PURPOSES.

BY

ROBERT GRIMSHAW, M.E., Etc.,

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TO THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS

PREFACE
TO FIRST EDITION OF PART II.

The matter in this volume is intended as supplementary to the Catechism, and needs no excuse for its appearance.

The extreme popularity and rapid sale of the Catechism proper warrants the author in putting in the same form, additional material of the same general character.

This volume contains the answers to many questions which were asked by engineers and steam users, and the answers to which do nôt appear in the Catechism itself.

It refers in many places to the Catechism, and is intended to be used with it.

Particular attention is paid to compound engines, to link motions, to the details of various existing types of steam engines, and to matters concerning setting up, running, adjusting, repairing, and altering particular builds of engines.

There is a probability that another supplementary volume may be issued in the course of a short time.

PREFACE
TO THIRD EDITION OF PART II.

In this edition the reader will find about 22 pages added, embracing about 70 questions and answers, mostly concerning the slide valve; while a number of typographical errors (none of them, however, important as affecting the sense) have been corrected. The Index, covering nineteen closely printed pages, will be found to contain the proper references to the new matter just added.

If there is any question relating to any matter concerning any kind of a steam engine, and which is not answered either in Part I. or in this volume, I should be very happy to receive it for reply by mail; and such questions and their answers will probably be added to future editions or volumes.

Nov. 1, 1887.

SUPPLEMENT

TO THE

STEAM ENGINE CATECHISM.

EXPANSION, &C.

Q. What is the advantage of an adjustable cut-off on marine engines?

A. There are times where the engineer can take advantage of smooth seas and favorable winds, and use less steam because the work is less.

Q. Is there any advantage of early cut-off on locomotives?

A. Mr. H. S. Hayward, Master Mechanic of the N. Y. Division of the Penn. R.R., says in a paper before the Am. Soc. Mech. Eng.:—

"In my railroad experience I find that a great deal of economy has also been realized in our locomotives by encouraging engineers to carry their steam up to the highest point allowed them, and cutting off their engines shorter instead of using the throttle."

Q. What is the disadvantage of very early cut-off on marine engines?

A. Where the cut-off is too early the engines are apt to race, and the wheel velocity to vary greatly.

Q. How may this be remedied?

A. This may be remedied in part by making the outer rim of paddle wheels very heavy. Mr. H. S. Hayward says, speaking of the ferry boat *Baltimore*, where the cut-off is adjustable and at times very early:—"We turn a very even wheel. We have constructed the outer ring of our wheel very much thicker than heretofore, and the boat is more free from the vibrations which you find on ferry boats."

Q. What is the disadvantage of too early cut-off on locomotives?

A. Wheel slippage, in many cases.

Q. On vertical engines should the cut-off be later on the upward or on the downward stroke?

A. On the down stroke.

Q. How is clearance volume often reckoned?

A. In parts of the stroke supposed to be multiplied by the piston area. Thus, if the

piston stroke is 20″ and area 200 square inches, and the clearance volume at one end is 200 cubic inches, then the clearance would be reckoned as equal to 200÷200=1″ of stroke.

Q. Supposing expansion perfect, under proper conditions, how may the "absolute" steam pressure in the cylinder, after cut-off, be ascertained?

A. Supposing expansion perfect, under proper conditions, the "absolute" steam pressure in the cylinder, after cut-off, should be just proportionate to the cylindrical volume bounded by the cylinder head at one end and the piston head at the other.

Assuming for the present that there is absolutely no clearance space between the piston head and the cylinder head when at either stroke end, and that cut-off takes place at one fourth stroke or 6″ in an engine having 24″ stroke, the terminal pressure above vacuum will be one fourth that at the point of cut-off; the pressure (*above vacuum*) when the piston has made half a stroke, or 12″ advance, will be half that at the point of cut-off; when it has made three fourths stroke, or 18″ advance, it will be one

third that at the cut-off, and so on. In other words, *the pressure above vacuum is inversely proportioned to the volume the steam occupies.*

Q. With cut-off at ¼, what initial pressure would be needed to get a mean effective pressure of ten (10) lbs., in a non-condensing engine, supposing there to be no clearance, cushion, wire drawing, throttling, leak, cylinder condensation, etc.

A. The average total pressure required would be 10+14.7=24.7 lbs.; the expansion rate 4, which has 1.3863 for its hyperbolic logarithm; then the total initial pre sure above vacuum would be 24.7÷(.25×1+1.3863)=24.7÷(.25×2.3863)=24.7÷.59658 =41.4 lbs.; and the initial pressure by the gauge would be 41.4—14.7=26.7 lbs.

Q. Suppose the foregoing conditions to be changed by a back pressure of 2 lbs., what initial pressure would be required?

A. Then the average total pressure required would be 10+2+14.7=26.7 lbs., and the initial total pressure 26.7÷.59658=44.7 lbs.; hence the initial pressure "by the gauge" would be 44.7—14.7=30 lbs.

Q. At what point in the stroke would

the cut-off have to take place, in the above mentioned case, to make the average effective pressure 30 lbs., instead of 10?

A. As the mean effective pressure can never be greater than the initial gauge pressure by the gauge in a non-condensing engine, and as the desired mean effective pressure is 30 and the initial gauge pressure only 26.7 lbs., the engine couldn't do the work.

Q. How could the work be done, then?

A. By adding a condenser.

Q. At what point in the stroke would the engine have to cut off-to give 30 lbs. mean effective pressure, with 26.7 lbs. initial pressure by the gauge, using a condenser?

A. Depends on the amount of vacuum. If the condenser gave a vacuum of only 5 lbs., then the engine would have to give 25 lbs. average pressure above atmosphere (39.7 lbs. av. total pressure) besides the work done by the condenser; and as the initial pressure is only 26.7 lbs., cut-off would have to be very late. In figures, — as the hyperbolic logarithm of the actual expansion rate is found only by dividing the

average total by the initial total pressure, and the quotient by the period of full steam plus the clearance, and then taking one from the quotient, we have two unknown quantities to work with, and the calculation will have to be made by approximation, or on the "cut and try" principle.

Assume cut-off at $\frac{9}{10}$, (expansion rate 1.111, the hyperbolic logarithm of which is 0.1044) we find that to give us an average total pressure, (with 41.4 initial total), of

$$41.4\ (0.9 \times 1.1044) =$$
$$41.4 \times .99396 = 41.14 \text{ lbs.},$$

which is too much.

Trying cut-off at $\frac{8}{10}$ (expansion rate 1.25, with hyperbolic logarithm .2231) we have that giving an average total pressure (with 41.4 initial) of

$$41.4\ (0.8 \times 1.2231) =$$
$$41.4 \times .97848 = 40.5$$

Trying cut-off at $\frac{7}{10}$ (actual expansion rate about 1.43, with hyperbolic logarithm 0.3577); that would give, with initial total of 41.4 lbs.,

$$41.4\ (0.7 \times 1.3577) =$$
$$41.4 \times .95039 = 39.35 \text{ lbs.},$$

which is less than the 39.7 which we want;

so we know that the cut-off required for 5 lbs. vacuum will be between 0.7 and 0.8 stroke.

For other amounts of vacuum the principle is the same.

Q. How is the work of the steam during admission (including overcoming atmospheric and other counter pressure) reckoned?

A. By multiplying the total pressure on the piston by the period of admission.

Q. How is the volume of steam admitted or used during the "full steam" period, measured?

A. It is equal to the product of the piston area by the period of admission plus the clearance.

Thus, if there is 2″ of clearance and cut-off takes place at 5″ of stroke, and the piston area is 260 square inches, the volume of steam is $260 \times (2+5) = 260 \times 7 = 1820$ cubic inches.

Q. How do you calculate the total work done *during* expansion to the end of the stroke?

A. Multiply the total pressure above vacuum on the piston area by the period of

admission plus the clearance, and by the hyperbolic logarithm of the actual expansion rate. (This includes the work done in overcoming all counter pressure above vacuum.)

Q. How do you find the initial pressure required for a given net quantity of work in one stroke?

A. To the net work expressed in foot pounds, add the continuous product of the piston area in square inches, the average back pressure in lbs. per square inch for the whole stroke, and the length of stroke in feet; divide the sum of these quantities by the piston area in square inches, and by a number got thus:—Add one to the hyperbolic logarithm of the actual expansion rate; multiply the sum by the period of admission plus the clearance (both in feet); and take the clearance in feet from the product.

As a formula:—

$$P = \frac{W + a\,p'\,L}{a\,[\,l'\,(1 \times \text{hyp. log } R') - c\,]}$$

Thus:—Suppose the given net work per single stroke to be 2,000 foot pounds; the

piston area 260 square inches; the average back pressure one pound per square inch by the gauge; the stroke 2′; cut-off at ½ stroke, and clearance equal to 0.1 of stroke; then we have—

2,000+(260×15.7×2) = 10,164 ft. lbs; 10,-164÷260 = 39.09; the actual expansion rate 2.1÷1.1=nearly 1.91, the hyperbolic logarithm of which is .6472; (1.1×1.6471)—0.1= 1.7118; 39.09÷1.7118 =23.83 lbs. absolute; 23.83—14.7 = 9.13 lbs. by the gauge.

(The engine would develop at 110 revolutions per minute 2,000×110×2 = 440,000 ft. lbs., or 440,000÷33,000 = 13.3 H. P.)

Q. How do you find the net work done by the steam for one stroke of the piston, with a given cut-off?

A. Find the actual expansion rate. Get its hyperbolic logarithm and add one thereto. Multiply the sum by the period of admission plus the clearance (in feet). From the product take the clearance. Multiply the remainder by the total initial pressure in lbs. per square inch. This gives the *total* work done, in foot pounds per square inch, on the piston. To get the negative work per square inch of back pressure in foot

pounds per square inch, multiply the average back pressure in pounds per square inch by the length of stroke in feet. Take the product from the total work per square inch, and this will give the net work per square inch. Multiply the remainder by the piston-area; this gives the net foot pounds for one stroke.

Q. How do you find the period of admission required for a given actual expansion ratio?

A. Multiply the clearance proportion by one less than the actual expansion ratio; take this from one; divide by the actual expansion ratio.

Thus:—Suppose 2′ stroke, actual expansion rate 3.5; clearance five per cent; then the period of admission is

$$\frac{1-[(3.5-1)\times.05]}{3.5}=\frac{1-.125}{3.5}=.25 \text{ stroke,}$$

(or cut off at six inches).

Q. How do you find the pressure at stroke end, or at any other point in the stroke after expansion, of steam expanded in the cylinder?

A. Divide the initial pressure above vacuum by the actual ratio of expansion

calculated to the given point of the stroke.

Or, multiply the initial pressure by the period of admission plus the clearance, and divide the product by the length of the part of stroke described up to the given point, plus the clearance.

Thus:—Suppose initial pressure sixty lbs. by the gauge, clearance five per cent, cut-off at one-fourth, required the pressure at $\frac{3}{4}$ stroke. The expansion rate *up* to *three-fourths stroke* is $(.75+.05)\div(.25+.05)=.80\div.30=2.667$ and $(60+14.7)\div2.667=74.7\div2.667=28$ lbs. absolute, or $28-14.7=13.3$ lbs. by the gauge.

$$\text{Or, } \frac{74.7\times(.25+.05)}{.75+.05}=\frac{74.7\times.30}{.80}=28 \text{ lbs.}$$

above vacuum, or 13.3 lbs. by the gauge.

Q. How do you find the net cylinder capacity for a given quantity of steam admitted for one stroke and a given actual expansion rate?

A. Multiply the volume of one lb. of steam at the given pressure (as found in a reliable table) by the given weight in pounds and by the actual expansion rate. Multiply

the result by 100 and divide by 100 plus the clearance percentage.

Thus:—Suppose we want to expand five lbs. of steam at 100 lbs. per square inch above vacuum at an actual expansion rate of 3.5, in a cylinder where clearance is 5%.

The volume of one lb. of steam at 100 lbs. absolute, is 4.33 cubic feet. Thus the net capacity is

$$\frac{5 \times 4.33 \times 3.5 \times 100}{1.05} = 72.167$$

cubic feet, or, say a cylinder of about 4 stroke and eighteen square feet of piston area, or having a diameter of about 4.8′ = 57.6″.

Q. How do you find the net cylinder capacity required to do a certain amount of total actual work in one stroke, with a given initial pressure and actual expansion ratio?

A. Divide the given work by the total actual work done by one lb. of steam of the same pressure and with the same actual expansion rate; this gives the weight of steam required, and from that can be found by a preceding rule the net capacity.

Q. How do you find the weight of steam admitted per cubic foot of the net cylinder capacity, for one stroke?

A. Divide 1 by the cylinder capacity per pound of steam.

Q. How do you determine the total actual work done per cubic foot of net capacity, for one stroke?

A. Divide 1 by the cylinder capacity per foot pound of work done.

Q. How do you find the total actual work done per square inch of piston?

A. The total amount of actual work done per square inch of piston is $\frac{1}{144}$ part of the work done per square foot. (See above.)

Q. What are the causes of differences between boiler and initial cylinder pressure?

A. The causes, commencing at the boiler, are friction in the steam pipe, resistance of the regulating throttle valve, resistance in the ports and passages, and disappearance of actual energy when the steam passing from the small area of the ports into the larger area of the cylinder, has its speed reduced.

Q. How does back pressure by the gauge vary?

the result by 100 and divide by 100 plus the clearance percentage.

Thus:—Suppose we want to expand five lbs. of steam at 100 lbs. per square inch above vacuum at an actual expansion rate of 3.5, in a cylinder where clearance is 5%.

The volume of one lb. of steam at 100 lbs. absolute, is 4.33 cubic feet. Thus the net capacity is

$$\frac{5 \times 4.33 \times 3.5 \times 100}{105} = 72.167$$

cubic feet, or, say a cylinder of about 4 stroke and eighteen square feet of piston area, or having a diameter of about 4.8′ = 57.6″.

Q. How do you find the net cylinder capacity required to do a certain amount of total actual work in one stroke, with a given initial pressure and actual expansion ratio?

A. Divide the given work by the total actual work done by one lb. of steam of the same pressure and with the same actual expansion ratio; this gives the weight of steam required, and from that can be found by a [illegible] rule the net capacity.

Q. How do you find the weight of steam admitted per cubic foot of the net cylinder capacity, for one stroke?

A. Divide 1 by the cylinder capacity per pound of steam.

Q. How do you determine the total actual work done per cubic foot of net capacity, for one stroke?

A. Divide 1 by the cylinder capacity per foot pound of work done.

Q. How do you find the total actual work done per square inch of piston?

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Q. How does back pressure by the gauge vary?

A. Back pressure by the gauge varies in non-condensing stationary engines and in locomotives nearly

As the square of the speed.

As the pressure of the steam at the instant of release (commencement of exhaust).

Inversely as the square of the area of the orifice of the exhaust nozzle.

It is less with high expansion rates than with low; and less with slow exhaust than rapid; less with dry than with wet steam, hence greater in unprotected than in well-covered cylinders (the proportion being 1.72 to 1 in observed cases).

It may be said for all engines, condensing or non-condensing, that "in the same engine, going at the same speed, the excess of the mean back pressure above the pressure of condensation, varies nearly as the density of the steam at the end of the expansion; and that in the same engine, with the same density of steam at the end of the forward stroke, that excess of back pressure varies nearly as the square of the speed."

Q. "Why does not the exhaust steam show a difference of temperature corresponding to the difference of power to be

obtained from steam at boiler pressure and exhaust steam?"

[The foregoing question was asked us in POWER, and was answered as follows :—]

A. The relation between the pressure on steam and its temperature is in a varying ratio, and the variation of this ratio is of a complicated character. The pressure of steam increases with the temperature, but the *rate* of increase of temperature increases as the pressure rises. Thus at 212° F. the pressure above vacuum is 14.7 lbs. per square inch, and at 213° the pressure rises 0.29 lbs.; at 247° the pressure above vacuum is 28.34 lbs. per square inch; and at 248° it has risen about ½ lb. At 432° the pressure is 350.73 and the rise in pressure per degree of temperature is 3.64 lbs., or about fourteen times as much rise of pressure per degree, as at atmospheric pressure. This relation between temperature and pressure is the same so long as the steam is in communication with unevaporated water.

THE SLIDE VALVE.

Q. What is a rule of thumb for lap of slide valve to cut off at two-thirds?

A. To cut off at two-thirds a well proportioned valve may have lap equal to one-fourth the stroke. (Chas. E. Emery.)

Q. What is the rule of thumb for inside lap of condensing engines?

A. One-half the outside lap.

Q. Which will stand the most inside lap, a condensing or a non-condensing engine?

A. The condensing engine.

Q. How may the evils of over cushion from excessive inside lap, generally be remedied in part?

A. By giving the valve more lead, so that the final compression is against the steam in the chest and not against the valve face.

Q. How should a slide valve be set so as to equalize the exhaust?

A. Put the crank on the front centre and mark coinciding centre-punch marks on cross-head and guide. Put the crank on the back centre, and centre-punch the guide again, to correspond with the cross-head mark. Move the eccentric until the valve has the lead for the forward stroke. Move the crank, in the direction the engine is to run, until the exhaust of the opposite

stroke closes; scribe a line on the guide to correspond with the cross-head mark; move the crank on until the other exhaust closes; scribe on the guide to correspond with the cross-head mark. If then the exhaust does not close at equal distances from the stroke ends, alter the length of the eccentric rod until it does; then bring the crank back to the dead centre, and shift the eccentric until the forward stroke has the required lead.

Q. How may you tell where the valve is at the time of closure, without opening the chest?

A. By a valve gauge of stout wire rod of this shape,

fitting centre punch-marks on the valve stem and on the stuffing-box (*not* on the gland).

Q. How can you tell when the valve has got lead, without opening the chest, if there are no punch marks to go by?

A. Put the crank on the centre, open the drip cock at that end, and then turn the eccentric ahead until steam shows at the drip cock.

Q. How do you find the correct length of the eccentric rod?

A. Mark the centre lines of cylinder, valve, rocker and main shaft. Put the crank on the back centre, and the valve so that the required lead is given for the forward stroke. Find the position of the eccentric centre by laying off the angular advance from the 90° line. Measure from this centre to the rocker pin (or to the valve stem pin if there is no rocker) for correct length of eccentric rod.

Q. If the lead opening is measured correctly for the forward stroke, will it be correct for the return stroke, if the eccentric rod length is measured as above directed?

A. Yes; because the opening is measured while the crank is on the central line—hence there is no irregularity of piston position owing to the angularity of the connecting rod.

Q. Is equalization of exhaust closure always possible?

A. Yes; because there may be added a slight amount of inside lap to whichever end of the valve needs it, so as to hasten exhaust closure on one stroke and delay it on the other.

Q. In equalizing cut-off should equalization of exhaust closure be considered, also?

A. No; because the only practical method for equalizing cut-off also equalizes the exhaust well enough.

Q. On what does the expediency of equalizing cut-off depend?

A. Upon that of giving unequal lead.

Q. Where is unequal lead disadvantageous?

A. For high speed engines.

Q. What alterations in the valve motion are requisite, in order to equalize cut-off?

A. Increase in angular advance, and lengthening the eccentric rod.

Q. In what class of engines is the steam pressure in the two ends more apt to differ; in vertical or in horizontal?

A. In vertical engines.

Q. Which end of a vertical engine generally gets the highest average pressure?

A. The lower.

Q. To reverse an engine having a single eccentric, what is necessary?

A. To turn the eccentric so that its centre line shall come on the other side of the centre line of the engine.

Q. In order to keep the lead opening and the lap the same for different positions of the eccentric for a reversing engine, what is necessary?

A. That the eccentric be slotted so that its centre can pass across the centre line of the engine in a direct straight line, instead of in an arc, as where the eccentric is simply turned on the shaft.

Q. How does the position of an adjustable eccentric for a reversing engine affect the cut-off?

A. The nearer mid-gear the earlier the cut-off, whether the eccentric centre be moved along a straight line (as where the eccentric is slotted) or in an arc, (as where it is simply turned on the shaft.)

Q. How does the position of the eccentric of a reversing engine affect the exhaust?

A. The nearer mid-gear the earlier the exhaust, whether the eccentric centre move in a straight line or in an arc.

Q. How does the position of the eccentric centre in a reversing engine affect the lead angle and the angular advance?

A. The nearer mid-gear, the greater the lead angle and angular advance, whether

the eccentric centre be moved in a straight line or in an arc.

Q. What is the lead opening at mid-gear?

A. At mid-gear the lead opening of the valve equals the port opening.

Q. What is the cut-off angle at mid-gear?

A. At mid-gear the cut-off angle equals the lead angle of the valve.

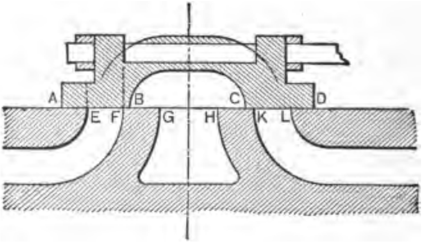
Q. How should valve lead be stated or reckoned?

A. Lead should be stated in degrees rather than in inches of either valve travel or piston motion.

Q. How should steam ports be proportioned?

A. After you have settled the throw of the eccentric, the angle of advance or angular lead, and the steam port width $EF=KL$, we might say as a general rule that the bridges FG, HK, should be wider than the distance at which the outer edge A of the valve, at its greatest distance from the central position, stands beyond the inner edge F of the admission port; otherwise the valve would blow through. The sum of the two

laps *AE* and *BF*, plus the port width *ES*, must be greater than half the valve travel; hence the bridge-width *EF* must be greater than half the valve travel, less the sum of *AE* and *BF*. You might say that the bridge-width *FG* should be equal to half the steam port width *EF* plus $\frac{1}{32}''$ to $\frac{1}{16}''$; and the exhaust port *GH*, equal to half the valve



travel plus the steam port width *EF*, plus the inside lap *BF*, less the bridge-width *FG*.

Q. In designing an Allen valve for an old seat, what is generally a good thing to do?

A. To widen the end ports in the seat by chamfering off the outside edges.

Q. What precaution must be taken in designing an Allen valve for an old seat?

A. Not to let the supplementary passage in the valve get over the end port in the seat.

Q. Will the Allen valve save fuel?

A. Sinclair says, "In very carefully conducted experiments made on the Boston and Albany R. R., to compare the performance of the Allen valve with an engine equipped with a common valve, it was found that the Allen valve effected a fuel saving of seven per cent."

Q. What difference should there be made in the seats in designing an Allen valve and an ordinary slide?

A. For the Allen, the steam ports and bridges should be a trifle wider than for the ordinary slide.

Q. What special care should be taken in casting an Allen valve?

A. To get the supplementary passage smooth inside, and the thin shell strong enough to stand the pressure.

Q. When is the speed of a slide valve highest?

A. The speed of a slide valve is greatest at mid-throw.

Q. How is the D valve kept to its seat?

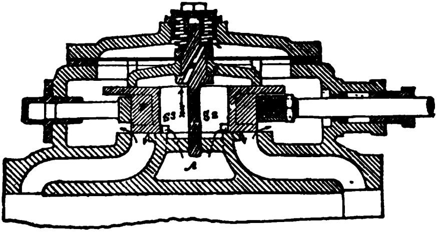
A. The D valve is kept to its seat by the

excess of the boiler pressure over that below the valve. The trouble is that it is too strongly held to its seat, causing undue friction, with its accompanying abrasion, and loss of power.

Q. What is one disadvantage of most balanced slide valves?

A. They do not lift if there is water in the cylinder.

Q. What is Tipping's balanced slide valve?



A. In this valve the steam is admitted at *A* inside the slide valve *S*, and is exhausted through the steam chest *C;* the valve *S* is open through, *B* is a back plate resting thereon nearly in equilibrium, *E* is a diaphragm plate, *F* are nuts, tightening which will relieve the pressure on the valve, *G* is a

spiral spring, the compression of which will add pressure in adjusting the valve. This arrangement affords a light, handy, and simple device, which we understand works very well, and may be constructed either rectangular or circular in shape and to any size, and for either single or double ports. In the cut the valve stem is not carried through the cover.

VALVE MOTIONS.

Q. How may valve motions be classified?

A. Valve motions may best be divided into those with fixed, and those with variable cut-offs; and the latter may have the cut-off variable either only while the engine is stationary, or while it is running. The latter, again, may be affected either "by hand" only, or "automatically." (This last is a point neglected by Zeuner in his celebrated standard treatise on valve-gears.)

Q. What is the Dodd motion?

A. The Dodd motion (1839) moves the eccentric centre directly ascross its shaft by two wedges with reversed points. It was improved by Dubs of Glasgow.

Q. What are the objections to the Dodd wedge motion?

A. It cannot be corrected for inequalities of cut-off and exhaust closure without introducing pernicious inequality of port opening at mid-gear.

Q. What is the link operated by two fixed eccentrics intended to imitate, in its effects?

A. A single movable eccentric.

Q. When and by whom were two fixed eccentrics first used for reversible gear?

A. In 1829, by Wm. T. James of New York, on a steam carriage.

Q. How many classes of link motions are there?

A. Four—

1. Shifting link.
2. Stationary link.
3. Allan.
4. Walschäert.

Q. Who invented the so called Stephenson shifting link, and when?

A. Howe, in 1843.

Q. Who invented the stationary link, and when?

A. Daniel Gooch, about 1845.

Q. Who invented the so-called Walschäert motion?

A. Heusinger von Waldegg.

Q, What are the advantages of the link and two fixed eccentrics over a single shiftable eccentric?

A. It can be adjusted for irregularities in cut-off and exhaust closure due to the angularity of the connecting rod.

Q. How can the link be suspended so as to equalize the cut-off?

A. So as to give it, on the forward stroke of a direct acting engine, more elevation, so as to neutralize the tendency of the connecting rod and to cause later cut-off on the forward stroke.

Q. What is the disadvantage of suspension so as to equalize the cut-off?

A. There is caused a certain amount of reciprocating *slip* of the link on the block at each revolution, causing lost motion and wear.

Q. In marine engines which is the more important—equalization of the cut-off or reduction of slip?

A. Reduction of slip.

Q. Where is slip least?

A. Slip is the least when the link block is nearest the point of suspension.

Q. What does this indicate?

A. That where you need a minimum slip at a certain point of suspension, the saddle stud should be placed as nearly as possible *over* such point.

Q. With a stationary link, does the lead change with changes in gear position?

A. No.

Q. Are there any shifting link motions in which the lead remains the same at all gears in both directions?

A. No, but many shifting link motions can be arranged so as to give constant lead for all gears *in one direction.*

Q. Where must the tumbling shaft be placed?

A. Either above or below the central line of motion, so that either eccentric rod shall strike it when the gear is shifted from full forward to full backward gear.

Q. Should the eccentric rod be curved?

A. No.

Q. What should be the hanger length?

A. The hanger should be long enough

not to conflict with the tumbling shaft arm in any gear. The latter should be as long as, or longer than the hanger.

Q. Suppose the boiler or other obstacle prevents the link being put in full gear back, what must be done?

A. Either the tumbling shaft must be put below the link motion or the rocker arm lengthened so as to depress the central line of the motion and the whole motion. In this case the rocker arm centre should be at an inclination so that the one driving the valve stem should be at right angles thereto, and the other one at right angles to the central line of motion.

Q. What is the influence of rocker arm length on the valve stem?

A. The longer the rocker arm, the less the vibration of the valve stem.

Q. How is the slip of the link-block affected by the rocker-arm length?

A. The longer the rocker-arm the less the slip of the link-block.

Q. Can the link motion be laid out at an inclination to the piston motion without affecting the link action?

A. Yes; so that the angular advance of

the eccentric is laid off from a line at right angles to the central line of the link motion.

Q. With "open" or "uncrossed" locomotive eccentric rods, how are the rods to be attached?

A. The "forward" rod with the top and the "backward" rod with the bottom of the link.

Q. With "crossed" eccentric rod on a locomotive, how should the eccentric rods be attached?

A. The "forward" rod to the bottom and the "backward" rod to the top of the link.

Q. How might the inequality of compression due to the eccentric rod obliquity be overcome?

A. The compression might be equalized by varying the inside lap of one end.

Q. When the valve has equal lap on both sides how may the cut-off be equalized?

A. Where the lap is equal on both sides the cut-off may be equalized by lengthening the valve stem, which has the same effect as lengthening the laps for the forward valve-stroke, and *vice versâ.*

Q. How can the inequality of the cut-

off which would be caused by eccentric rod obliquity, be neutralized?

A. Unequal cut-off by reason of eccentric rod obliquity may be neutralized by giving the valve different steam lap on the two ends; but this would give unequal steam lead.

Q. How can the effect of the obliquity of the eccentric rod be neutralized?

A. The effect of the obliquity of the eccentric rod can be neutralized by lengthening the valve stem, so as to cause the greatest port opening at fore-stroke to be rather less than on the return; setting the valve "by the lead," that is, with the eccentric rod in its inclined position.

Q. When does the obliquity of the connecting rod most affect the position of the piston?

A. The obliquity of the connecting rod affects the position of the piston most when the crank is at right angles.

Q. At mid-gear what is the angular advance of the valve?

A. Ninety degrees (90°) of arc.

Q. At mid-gear where does the exhaust close?

A. At mid-gear it closes at half stroke.

Q. Under what circumstances is steam not admitted to the cylinder for either stroke, at mid-gear?

A. When the port opening is 0 at mid gear. (Then the lead opening is 0 and the lead angle 90°.)

Q. Is port opening reduced or increased towards mid-gear?

A. Decreased.

Q. Does the lead angle increase or decrease from full gear to mid-gear?

A. It increases, whether the lead *opening* remains constant or varies.

Q. With the Stephenson link motion and uncrossed eccentric rods, where is the lead least?

A. With the Stephenson link motion and uncrossed eccentric rods, the lead is least at full stroke.

Q. With the Stephenson link motion and uncrossed eccentric rods, where is the port opening least?

A. At mid-gear.

Q. With a "crossed rod" link motion, where is the lead least?

A. At mid-gear.

Q. With a "crossed rod" link motion, is the port opening less or greater at intermediate gear, than with the uncrossed rods?

A. Less than with the uncrossed rods.

Q. What should be the length of the link?

A. The link slot may have an available length of six times the throw of the eccentrics.

Q. What causes affect the variation of the lead at different grades with Stephenson link motion?

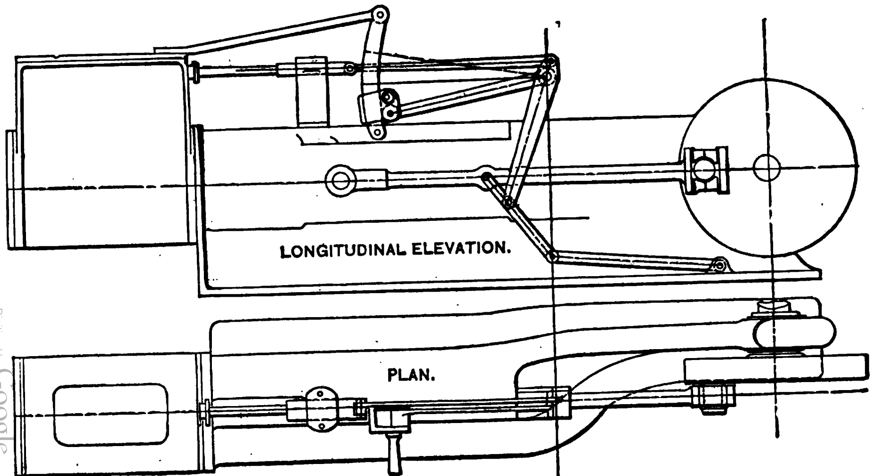
A. The variation of the lead at different grades, with the Stephenson link motion, is greatest with short eccentric rods and with long links.

Q. When a loose eccentric has lead, how far apart are its positions of forward and backward gear?

A. When a loose eccentric has lead, its positions of forward and backward gear have an angular distance equal to 180° less twice the angle of lead.

Q. When a loose eccentric has no lead, how far apart are its positions of forward and backward gear?

A. When a loose eccentric has no lead,



JOY VALVE GEAR. SWORD ARM DESIGN.

its forward and backward gear positions are 180° apart.

Q. What is the Marshall gear?

A. In the Marshall gear, the eccentric rod bears or constitutes a link, its free end traveling in a slot in a disc which may be changed in its position. The valve stem is drawn at right angles to the centre line of the eccentric rod or link.

Q. What is the Joy gear?

A. The Joy gear drives from the connecting rod, using no eccentric. We give illustrations of three different designs of this gear.

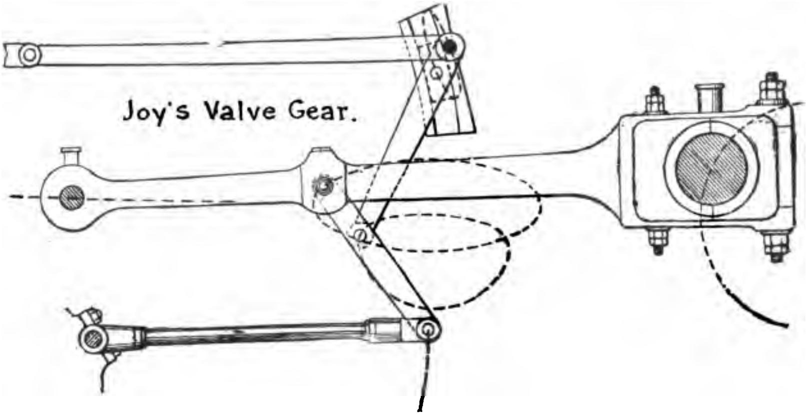
Q. What is the Brown gear?

A. The gear of Chas. Brown of Winterthur, dispenses with the eccentric, and the link is attached to the connecting rod.

THE COMPOUND ENGINE.

Q. What is a compound engine?

A. A compound engine is one in which, instead of the steam completing its expansion in the same cylinder in which it receives it from the boiler, it is only partly expanded in that cylinder, and the expansionis continued or completed in one or more cylinders (called low pressure cylinders) which



SLIDING QUADRANT DESIGN.

receive the exhaust of the first (or high pressure) cylinder. Sometimes one high pressure serves equally two low pressure cylinders in which the expansion commenced in the first is continued to an equal degree one with the other. Sometimes the expansion begun in one cylinder is continued in the second and completed in a third.

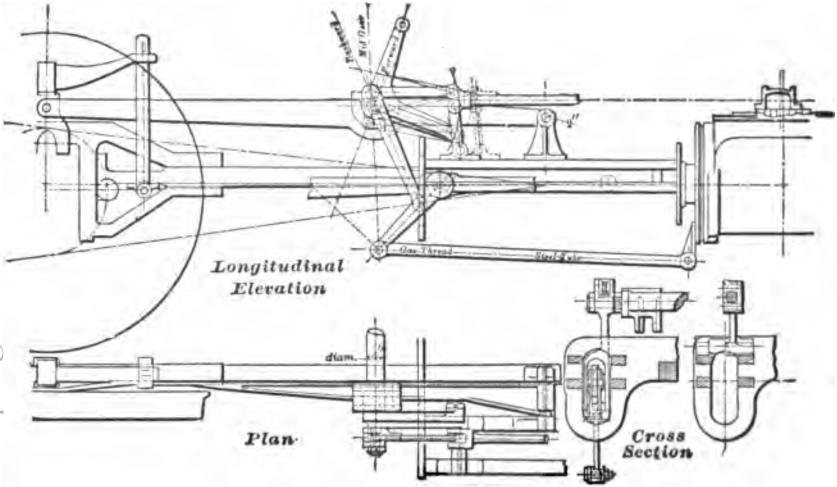
Q. For a compound condensing engine to exert the same power as a single cylinder condensing engine with the same number of expansions in each case, what would have to be the diameter of the condensing or low pressure cylinder of the compound engine?

A. Equal to that of the single condensing cylinder.

Q. Then, if the power could be got from one cylinder with the steam expanded a certain number of times, would it pay to make the more expensive compound engine with two cylinders to expand the same number of times?

A. No.

Q. Where then has the compound condensing engine any advantage over the single cylinder condensing engine?



JOY VALVE GEAR. SWING LINK DESIGN.

A. The steam at high temperature and pressure entering the high pressure cylinder of a compound engine is not chilled by the low temperature of the cylinder due to the low terminal pressure; also the thrusts of the compound cylinders on the crank pins and crank shaft are more uniformly distributed around the axis than where there is but one cylinder.

Q. Who invented the principle of the compound engine, and when?

A. Jonathan Hornblower patented it in 1781, but could not apply it because Watt's patents were in existence.

Q. Who first applied the principle of the compound engine, and when?

A. Arthur Woolf patented in 1804 the earliest compound engine in which the principle was practically carried out.

Q. In Woolf's early engines, how were the cylinders placed?

A. Both cylinders were at one end of a working beam; the condensing cylinder at the outer end, and the high pressure cylinder closer up, with less stroke.

Q. What was the principal objection to this Woolf engine?

A. All the power was applied on one side of the centre of the beam.

Q. How and when was this fault first remedied, and by whom?

A. By Wm. McNaught, in 1845, putting the condensing cylinder at one end of the beam and the high pressure cylinder between the main centre and the crank.

Q. What is another good arrangement of a Woolf engine?

A. By making the cylinders horizontal and side by side, secured to a single bed plate; both piston rods having one crosshead in common, one connecting rod taking the whole power to the crank.

Q. What is the best proportion between the high and low-pressure cylinders?

A. Turnbull says: It has been found from modern practice that when the length of stroke of both cylinders is the same, it is necessary that the condensing cylinder be about three times greater in area than the high-pressure one, and this proportion is best suited when the steam employed is from 45 to 50 lbs. pressure above the atmosphere, and cutting off the steam after being admitted during one-third of the stroke in

A TABLE SHOWING THE RELATIVE AREAS OF THE TWO CYLINDERS OF A COMPOUND ENGINE, WITH THE AVERAGE PRESSURE IN EACH CYLINDER, ETC.

P	R	P′	H	S	p′	p
30	2.449	12.25	.896	23.23	7.52	15.71
35	2.645	13.22	.972	26.10	7.83	18.27
40	2.828	14.14	1.040	28.85	8.04	20.81
45	3.000	15.00	1.098	31.47	8.22	23.25
50	3.162	15.86	1.150	34.00	8.40	25.60
55	3.316	16.58	1.197	36.44	8.58	27.86
60	3.464	17.32	1.242	38.83	8.74	30.09
65	3.605	18.02	1.281	41.13	8.85	32.28
70	3.741	18.70	1.319	43.39	9.08	34.31
75	3.872	19.36	1.353	45.57	9.11	36 46
80	4.000	20.00	1.386	47.72	9.24	38.48
85	4.123	20.61	1.415	49.78	9.34	40.44
90	4.242	21.21	1.444	51.85	9.44	42.41
95	4.358	21.80	1.470	53.84	9.56	44.28
100	4.472	22.36	1.497	55.84	9 64	46.20
105	4.582	22.91	1.521	57.77	9.73	48.04
110	4.690	23.45	1.545	59.69	9.82	49.87
115	4.795	23.98	1.567	61.57	9.89	51.68
120	4.898	24.45	1.589	63.43	9.96	53.47
125	5.000	25.00	1.609	65.32	10.05	55.27
130	5.099	25.50	1.629	67.02	10.13	56.89
135	5.196	26.00	1.647	68.77	10.21	58.66
140	5.291	26.46	1.665	70.51	10.26	60.25
145	5.385	26.93	1.683	72.25	10.32	61.93
150	5.477	27.38	1.700	73.95	10.38	63.57
155	5.567	27.84	1.716	75.67	10.46	65.21
160	5.656	28.32	1.732	77.28	10.52	66.76
165	5.744	28.72	1.748	78.93	10.58	68.35
170	5.830	29.15	1.763	80.56	10.64	69.92
175	5.916	29.58	1.777	82.14	10.70	71.44
180	6.000	30.00	1.791	83.73	10.75	72.98
185	6.082	30.41	1.805	85.32	10.80	74.52
190	6.164	30.82	1.818	86.86	10.85	76.91

the high-pressure cylinder. When the steam to be employed is of a less pressure, but the point of cut-off the same, then the relative proportions of the cylinders must be nearer to each other, and the reverse when steam of a greater pressure is to be used. The table on page 49 is prepared on a basis which accords with correct theory and good practice:—

The first column $=P=$ initial pressure of the steam above a perfect vacuum on entering small cylinder; the second $= R = \sqrt{\frac{P}{t}}$ shows the relative areas of the two cylinders, and also the number of expansions in high-pressure cylinder; the third column, $=P'=$ the terminal pressure in high-pressure cylinder, gives the pressure at beginning of stroke in condensing cylinder; the fourth column, $= H$, contains the hyperbolic logarithms of R; the fifth, $= S$, gives the average pressure during a stroke in a single cylinder, for the different values of R and $= P\frac{1 + H}{R}$; the sixth column, $=p'$, gives the average pressure during a stroke in the con-

densing cylinder of a compound engine, and $=P'\frac{H}{R-1}$; and the last column, p, gives the average pressure during a stroke in the high-pressure cylinder $P=\frac{1+H}{R}-P'\frac{H}{R-1}$.

Q. How many principal forms of compound engine are there?

A. Two; one where the exhaust from the first cylinder goes direct into the second (Woolf type), and one where there is an intermediate receiver.

Q. How many kinds of piston movements are possible in receiver compound engines?

A. Three—

1. Entire independence.

2. Equal number of strokes per minute, with initial points in certain relation, and

3. Further limited by periods of tarrying (as in the Worthington pump).

Q. How do the pistons move in the Woolf type?

A. Together. The steam from the top of the first cylinder exhausts into the bottom of the second, and *vice versâ*.

Q. How do the pistons of a receiver engine move?

A. In a receiver engine, the pistons are connected to cranks on one shaft, at right angles to each other.

Q. Comparing the Woolf and the receiver types of compound engine, which is the more economical of steam?

A. When there is no clearance and no intermediate fall of pressure, there is no difference in the steam economy of the Woolf and the receiver types, but when there is an intermediate fall of pressure, the work done on the receiver system is greater than by the Woolf type.

Q. In a compound engine, how is the actual rate of expansion in the first cylinder, found?

A. To find the actual rate of expansion in the first cylinder, divide the period of admission plus the clearance, into the length of stroke plus the clearance. Thus, suppose cut-off at $\frac{1}{3}$ and clearances at one end equal to $\frac{1}{20}$ the piston displacement, then the actual expansion rate would be, (just as in an ordinary non-compound engine,)

$$1.05 \div 0.25 = 4.2.$$

As a formula, $R' = \frac{L'}{l'}$.

Q. How do you find the ratio of intermediate expansion in a compound engine (receiver type)?

A. The ratio of intermediate expansion may be got from the pressure before and after expansion. Thus, if the fall of pressure was from 20 lbs. (*absolute*) in the first cylinder, to 15 lbs. in the second, the ratio would be $20 \div 15 = 1.333$.

Q. Does the volume of the receiver give any evidence as to the expansion between the first and the second cylinders?

A. No.

Q. How may we determine the rate of ntermediate expansion between first and second cylinders, in a Woolf engine?

A. Either by the method necessary in the case of a receiver engine, or from the volume of the intermediate space.

Q. How do you find the ratio of expansion in the second cylinder in a Woolf engine?

A. Find the ratio of the capacity of the first cylinder, plus its clearance, plus the

intermediate space, to the capacity of the second cylinder plus the intermediate space, (taking this last to include the clearance of the second cylinder).

Q. Is the ratio of expansion in the second cylinder of a receiver engine affected by clearance?

A. No.

Q. How do you find the actual expansion rate in the second cylinder of a receiver engine?

A. Where there is no intermediate fall of pressure, the actual expansion rate in the second cylinder is the ratio of the cylinder volumes. Where there is an intermediate fall of pressure, the ratio of actual expansion in the second cylinder is reduced. Thus, if the cylinder ratio is 3, and the ratio of the intermediate fall to the final pressure in the first cylinder is 4, then the actual expansion rate in the second cylinder will be $(4-1) \times 3 \div 4 = 2.25$.

Q. How do you find the total actual expansion rate for a Woolf engine?

A. Multiply the ratio of the first to the second cylinder by the length of stroke, and divide by this length plus the clear-

ance; add to the quotient the ratio-value of the intermediate space; multiply this sum by the actual ratio of expansion in the first cylinder. Thus, suppose cut-off in the first cylinder at $\frac{1}{4}$; clearance $\frac{1}{20}$; cylinder ratio 4; ratio value of the intermediate space $\frac{1}{3}$. Then the actual expansion rate in the first cylinder will be $1.05 \div 0.25 = 4.2$. The modified rate of the cylinders is $4 \times \frac{1}{1.05} = 3.809$; and $3.809 + .333 = 4.142$. Finally, $4.2 \times 4.142 = 17.396$ = total actual ratio of expansion.

Q. How do you find the combined actual ratio of expansion behind the pistons, in the Woolf engine?

A. Multiply the actual ratio of expansion in the first cylinder by the ratio of expansion in the second cylinder. Or divide the product representing the total actual ratio of expansion by the ratio-value of the intermediate space, plus one. Thus, if the total actual ratio of expansion is 17.396, and the ratio-value of the intermediate space is $\frac{1}{3}$, then $17.396 \div 1.333 = 13.05$, the combined actual ratio of expansion behind the pistons (in a Woolf engine.)

Q. How do you find the total actual expansion-ratio in a receiver engine?

A. Multiply the ratio of the first and second cylinder volumes by the actual expansion rate in the first cylinder. This gives the total actual expansion rate in a receiver engine.

Q. How do you find the combined actual ratios of expansion behind the pistons of a receiver engine?

A. Multiply the actual expansion rate in the first cylinder by the expansion rate in second cylinder. That is, multiply the ratio of the first and second cylinder volumes by the actual ratio of expansion in the first cylinder and by one less than the ratio of the intermediate fall of pressure to the final pressure in the first cylinder.

Q. How do you find the net work done in the two cylinders of Woolf engines, for one stroke, with a given combined actual ratio of expansion?

A. Add one to the hyperbolic logarithm* of the given combined actual ratio of expansion; multiply the sum by the period of admission to the first cylinder, in feet, plus the clearance, expressed in feet, of stroke; from the product subtract the clearance;

* See table of hyperbolic logarithms, p. 141 of Part I.

multiply the remainder by the net piston area of the first piston in square inches, and by the initial cylinder pressure in lbs. per square inch. This gives the net work in foot lbs. *per stroke.*

Thus :—suppose the net area of the first piston, 500 square inches ; of the second, 1,500; initial pressure, 63 lbs. (*absolute*); cut off at 2′ of the stroke, of 6′ clearance volume at one end, equal to 7%.

The actual ratio of expansion in the first cylinder is $6.42 \div 2.42 = 2.653$. The ratio of intermediate expansion we assume as 1.333. The actual expansion rate in the second cylinder is equal to the ratio of the capacity of the first cylinder plus its clearance plus the intermediate space, to the capacity of the second cylinder plus the intermediate space; this last including the clearance of the second cylinder ; or

$$\left(3 \times \frac{6}{6.42}\right) + .333 \div 1.333 = 2.353.$$

The modified ratio of the cylinders is $\times 5 \div 6.42 = 2.804$; and $2.804 + .333 = 3.137$, which multiplied by $2.653 = 8.322$, the total actual ratio of expansion. The combined

ratio of expansion behind the pistons is $8.322 \div 1.333 = 6.242$, the hyperbolic logarithm of which, as found in the proper table, is 1.8310.

Then the net work in foot lbs., during one single stroke, is $(1+1.8310)=2.8310$; $2.8310 \times 2.42 = 6.851$; $6.851 - 0.42 = 6.431$; $6.431 \times 63 \times 500 = 202{,}576.5$ foot lbs.

As a formula:—

$$w = aP[l'(1 + \text{hyp. log. } R'') - c]$$

Q. How do you calculate the net work done in the two cylinders of a receiver engine for one stroke, with a given actual ratio of expansion in the first cylinder?

A. Multiply the first cylinder's actual expansion rate by the cylinder ratio; and to the compound ratio of the two cylinders add one; multiply the sum by the initial period of admission into the first cylinder, plus the clearance in feet. Call this product (A). Divide one less than the ratio of the two cylinders by the actual expansion rate in the first cylinder. Add one to the quotient, and multiply the sum by the initial clearance in feet. Call this product (B). Subtract product B from product A. Call the remainder (C). Multiply the net

piston area of first cylinder in square inches by the total initial pressure in lbs. per square inch, *above vacuum*, and by the remainder *C*. The product is the net work in foot lbs. for one single stroke.

Thus ;—Suppose net area of first piston 500 square inches ; of second, 1,500 ; total initial pressure above vacuum, 63 lbs. per square inch; clearance=0.42′; cut-off at 2′; stroke 6′ ; then actual expansion rate in first cylinder=6.42÷2.42=2.653 ; second expansion ratio 3 ; compound ratio 2.653×3=7.959, the hyperbolic logarithm of which is 2.0743 (as found in table;) 3.0743×2.42=7.44=*A;* (3—1)÷2.653= .7162; 1.7162×.42=.737=*B;* 7.44—.737= 6.703 = *C;* 63×500×6.703 = 211,145 foot lbs., the net work during one single stroke.

Q. What is the difference between a Woolf and a Wolff compound engine?

A. The Woolf compound engine has no receiver ; the Wolff has a receiver.

DRYNESS OF STEAM.

Q. Is saturated steam at the condensing point or at the generating point ?

A. Saturated steam standing over water,

s at both the condensing point and the generating point.

Q. When does steam approach to the condition of a perfect gas?

A. When superheated.*

Q. What are the advantages of superheating steam?

A. The advantages of superheating are (1) To increase the efficiency of the steam without producing a dangerous pressure; (2) To "improve the vacuum," that is, to lessen the density required to overcome a given resistance, and lower the back pressure; (3) To prevent condensation during expansion.

Q. How far is superheating carried with ease?

A. To 100° above the normal steam temperature due to the pressure.

Q. What can be said about the money loss by radiation from cylinders?

A. Radiation of heat from uncovered or unjacketed cylinders walks off with profits, and most inexcusably; for there are few makes of engines which cannot be cheaply covered with a non-conducting lag-

* See pages 87 to 90 of Part I.

ging, in such a way as that all parts can be readily disconnected and got at. Felt and other lagging materials are plenty in the market; and air-space, which is very desirable between even the best lagging and the cylinders, costs nothing.

Q. Are steam traps any use in giving drier steam in the cylinder?

A. Steam traps appear to some to be a needless expenditure. But we notice that any one who has had a cylinder head knocked out by reason of water in the cylinder generally believes in dry steam, and in steam traps, ever thereafter.

Q. What advantage has dry steam over wet, as regards friction?

A. Dry steam gives less friction in steam and exhaust pipes and passages, than wet; hence gives higher initial pressure and less back pressure.

Q. How much "primage" may there be in steam?

A. Hirn's experiments show an average of five per cent; Zeuner states it as seven and a half to fifteen per cent.

Q. What is "commercially dry" steam?

A. Steam containing only three to fou

per cent of water cannot be distinguished, by observation, from that which is perfectly dry, and may therefore be said to be "commercially dry."

THE GOVERNOR.

Q. Which requires the more sensitive governor, a throttling engine, or an automatic; and why?

A. The automatic, because the pressure in the cylinder varies more during the stroke than in a throttling engine.

Q. Which needs the more sensitive governor, an early cut-off, or a late cut-off; and why?

A. The early cut-off, because of the greater variation in pressure during a stroke.

Q. What is the objection to the ordinary centrifugal speed-governor?

A. The trouble with the ordinary centrifugal speed-governor is that it "has to go slow in order to go fast." This is true of any speed-governor; it does not operate to check the speed until the engine has actually speeded up beyond the regulation rate. As a consequence, it takes a turn or two of running at abnormal rate, before the

engine can be brought down to regular speed. Now, suppose an engine running 120 turns and belted on to a "dynamo" making 1,200 turns; the engine may make one turn at the rate of 150 turns per minute, and then be choked down to one turn at the rate of 90, and then be speeded up to one at 140, then at 100, then at 130, then 110, and then settle down to 120, as if nothing had happened. A speed indicator applied during ten seconds would indicate an average speed of 120 turns, and the engine would be supposed to be a marvel of steady running; but the dynamo would have been running at rates from 1,500 down to 900, if the belt slip did not prevent some of these variations.

Q. How might the objection to the ordinary centrifugal speed-governor be remedied?

A. What is wanted is a governor which will govern by the load and not by the speed, and will govern so quickly that if the engine is speeded at 120, *each turn* shall be made in one-half second; no spurts, no lags, no dancing.

This can be accomplished in the case of

an engine driving a dynamo, by "shunting off" a part of the current, through a coil passing around a vertical soft iron bar attached to the throttle valve or cut-off lever. Any attempted increase of speed would increase the current in the coil, and raise the bar in the coil, thus actuating the throttle or the cut-off, in the veriest fraction of a second.

Q. What classes of work require the best governors?

A. Electric light machines, textile machinery (particularly for silk), flour mills, and saw mills.

Q. What law governs the revolution of the ordinary rotating conical pendulum governor?

A. One revolution is performed in the same time as two vibrations of a common oscillating pendulum, having a length equal to the height of the point of suspension of the conical pendulum above the plane in which the balls revolve.

Q. How long an oscillating pendulum will make a vibration in one second?

A. 39.1393″ in the latitude of London.

Q. How long an oscillating pendulum

would vibrate in $\frac{1}{70}$th of a minute in the latitude of London?

A. $$\left(\frac{\sqrt{39.193} \times 60}{70}\right)^2 = 28.75''.$$

Q. How do you determine the number of revolutions that a conical pendulum governor will make, when the balls are at a given distance below the suspension point?

A. Multiply the square root of the height of the cone in inches by 0.31986, and this gives the time of revolution in seconds.

Q. How do you determine the diameter of the circle described by the balls of a conical pendulum?

A. Divide 187.58 by the number of revolutions per minute, and the square of the quotient will be the distance in inches, of the plane of revolution, below the point of suspension. Take the square of this vertical distance from the square of the length of arm in inches, and the square root of the remainder is the radius of the circle in which the centres of the balls revolve.

Q. Has the weight of the governor balls any influence upon the governor speed?

A. No.

Q. What may be said about the governor speed?

A. The governor should in no case make a number of revolutions per minute, exactly a multiple of the rotation speed of the engine, as that causes "dancing," and "plays hob" in mills where regular speed is a necessity.

Q. In guaranteeing the regulation of a steam engine, what should be specified?

A. In guaranteeing the regulation of an engine, the extremes and the averages of variation of pressure, speed and load, should be specified.

For instance, an engine may be rated to run within 2% of 120 revolutions, provided the governor is never set to keep the speed below 110 or above 135; and the pressure is kept between 75 and 85 lbs., averaging 78; under a load averaging 65 H. P., and never running below 55 nor over 75 H. P.

DUTY OR ECONOMY OF AN ENGINE.

Q. What is properly meant by the "duty" of an engine?

A. The amount of work done, in relation to the steam consumed; though it is

generally understood as meaning the amount of work done, in relation to the fuel consumed by the boiler supplying the steam.

Q. What is the only means by which steam can do useful mechanical work?

A. By changing the volume of the vessel containing it.

Q. How is the efficiency of steam in a perfect engine reckoned?

A. By dividing the range of temperature worked through, by the maximum initial absolute temperature of the fluid entering the cylinders.

Q. How is the efficiency of an engine as a machine reckoned?

A. By the ratio of the quantity of work transmitted from the engine to the machinery of transmission, to the work done by the steam in the piston.

Q. How should the economy of an engine be expressed, and why?

A. The economy of an engine should be expressed in lbs. of dry steam per hour per H. P., because expressing economy in lbs. of fuel per hour per H. P. depends on the boiler too, rather than on the engine

alone. Thus one boiler might require 600 lbs. of coal to evaporate 3,600 lbs. of water from 62° to 310° F., and produce say 120 H. P. in a certain engine; another boiler might require only 335 lbs. of coal to produce the same amount of steam and power.

Q. In buying an engine, when the rating is given in lbs. of steam per hour, what should be specified?

A. In rating an engine in lbs. of steam per hour, it should be specified that the steam should be either saturated, dry or superheated*; and the initial pressure, point of cut-off, and load should be stated.

Q. Under what conditions does insufficient expansion cause waste?

A. Insufficient expansion is an evil which, especially in ordinary slide-valve engines cutting off at ⅝ stroke and later, is responsible for a great deal of the waste. The higher the terminal pressure, as compared with the mean effective pressure,* the greater the loss from this source.

Q. How far should expansion be carried?

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* See under these heads in Steam Engine Catechism, Part I.

A. In what is conventionally known as a "perfect" engine (in which the waste is *only* about two-thirds the entire amount of heat), expansion should be carried on until, in the case of a condensing engine, the terminal pressure would be as low as that corresponding to the temperature of the condensers. In the average non-condensing engine, we may say practically that the best economy of steam is obtained by expanding down until the terminal pressure is just as low as the back pressure.‡ Thus, roughly, if the back pressure is 2 lbs. above the atmosphere=16.7 lbs. absolute,§ and the initial pressure¶ in the cylinder was 68.8 lbs. by the gauge=83.5 lbs. absolute, the actual expansion rate‖ should be at least 83.5÷16.7=5; that is, not reckoning clearance, cut-off at least as early as one-fifth stroke. The steam user may readily prevent and cure this source of waste.

Q. Does expansion pay when there is a good vacuum?

A. The better the vacuum the greater the gain by expansion.

Q. Does expansion pay when there is a great deal of clearance?

A. The greater the clearance, the less the proportionate gain by cut-off at a fixed point of the stroke; because clearance practically lessens the expansion ratio.

Q. How far should expansion be carried?

A. In what is conventionally known as a "perfect" engine (in which the waste is *only* about two-thirds the entire amount of heat), expansion should be carried on until, in the case of a condensing engine, the terminal pressure would be as low as that corresponding to the temperature of the condensers. In the average non-condensing engine, we may say practically that the best economy of steam is obtained by expanding down until the terminal pressure is just as low as the back pressure. Thus, roughly, if the back pressure is 2 lbs. above the atmosphere=16.7 lbs. absolute, and the initial pressure in the cylinder was 68.8 lbs. by the gauge=83.5 lbs. absolute, the actual expansion rate should be at least 83.5÷16.7=5; that is, not reckoning clearance, cut-off at least as early as one-fifth stroke. The steam user may readily prevent and cure waste from a wrong grade of expansion.

Q. How much gain is there in adding lap so as to cut-off at half stroke?

A. If we suppose a theoretically perfect condensing engine, having no clearance, throttling, wire-drawing, *cylinder* condensation, nor steam or exhaust lead, and a perfect vacuum, we would get a theoretical

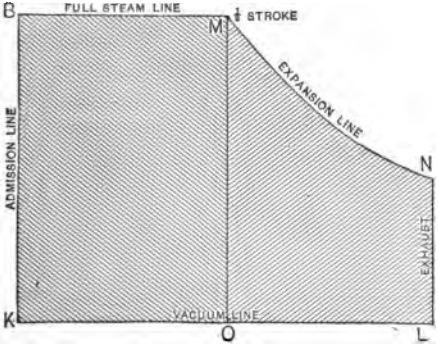

Fig. 6.

card like Fig. 6. when cut-off is at half stroke, and instantaneous. There would be an instantaneous rise of pressure, from perfect vacuum at K, to full boiler pressure at B; then the boiler pressure line would run at a uniform height, B M, until cut-off took

place (at half stroke in this case). At that point the pressure above vacuum would fall at such a rate that at stroke end it would be only half what it had been between commencement of stroke and point of cut-off. Then it would at once fall to vacuum, and remain at that until the end of the return stroke.

Such a diagram would be represented by the figure K B M N L; the height of the figure above the line K L representing pressures. The pressure at commencement of stroke is represented by B K; that at half-stroke, where cut-off takes place, by M O; that at stroke end, L N. The areas of the figures K M B O and O M N L represent the amount of work done before and after expansion, respectively. In this case it will be found that O M N L is 0.69 (or nearly seven-tenths) the area of K B M O; and that is the amount gained by cutting off at half stroke, *under the conditions named.*

Suppose that there was not a perfect vacuum, then instead of the pressure during the return stroke being nothing, it would be an appreciable quantity; and the

amount of work done would, instead of being K B M N L, be only A B M N D—as shaded. Of this, the part A B M R represents the work done during full steam, and the part R M N D, that done during expansion. (See Fig. 7.)

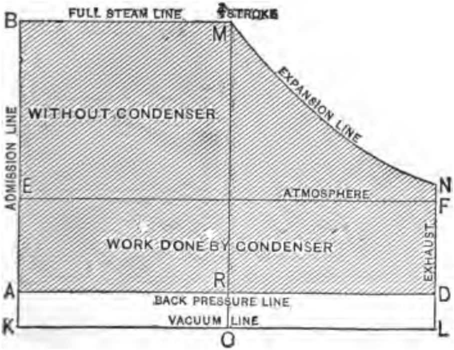

Fig. 7

In this case, the pressures above vacuum are the same as in Fig. 6, but the gain during expansion is not proportionately so great.

The better the vacuum, the greater the proportionate gain by expansion.

Now, suppose that this last case was altered by the introduction of clearance, say 2½%, (as shown in Fig. 8).

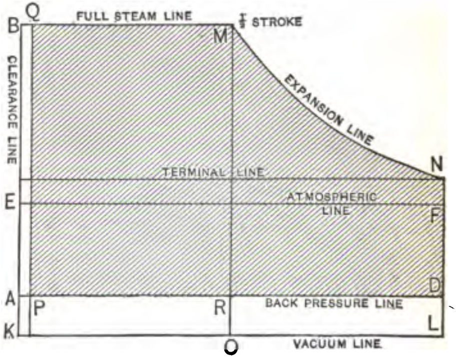

Fig. 8.

If Q M or P R is half of the whole stroke P D, then B M O K will be more than half of the whole volume, and N L will be more than half the initial pressure, K B. We have plainly marked all the lines of this diagram. It will be seen that the work done during expansion, (represented by the area R M N D) is not quite as large a proportion of the work done during full

steam, represented by the area Q M R P, as if there had been no clearance.

The greater the clearance, the less the proportionate gain by cut off at a fixed point of the stroke; because clearance practically lessens the expansion ratio.

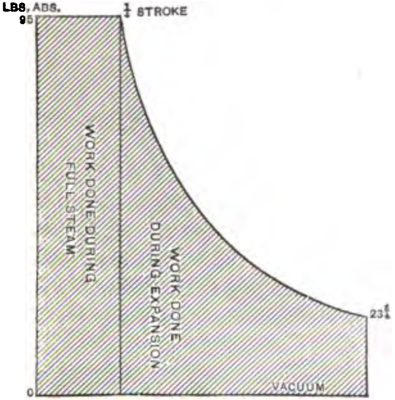

Fig. 9.

I will supplement my answer to the above question by showing what the proportions would be between the work done during full steam and that done during expansion, supposing all the conditions

to be as in Fig. 6, except that cut-off is at one-fourth stroke instead of at one-half. In this case there is much more work done during expansion than before (see Fig. 9).

The earlier the cut-off, under the conditions named, the greater the economy of steam by expansion.

But economy of steam is not necessarily economy of money.

Q. Is it not generally admitted that ¼ stroke is the most economical point of cut-off?

A. That depends upon the conditions. For instance, with initial pressure 27 lbs. by the gauge and cut-off at ¼, in a non-condensing engine, expansion would go below the atmospheric line, and during about the last third of the stroke the piston would drag on the crank pin.

Q. How much economy is there in high initial pressures?

A. I clip from my note book the following table showing the decreased steam consumption, consequent on increased initial pressure at such a point of cut-off as to give the stated mean effective pressures:

INITIAL PRESSURE.	M. E. P.	Lbs. Water per Hour per H. P.	Gain of Power per Increment of 10 Lbs.	Economy of Fuel per Increment of 10 Lbs.
10 Gauge Lbs.....	9.3	75.3	per ct.	per ct.
20 "	16.3	42.9	76	43
30 "	21.2	33.	30	23
40 "	25.6	27.3	20	17
50 "	29.1	24.	14	12
60 "	32.	21.9	10	9
70 "	34.5	20.3	8	7
80 "	36.3	19.2	6	5
90 "	38.1	18.4	5	4
100 "	39.2	17.8	3	3

Q. Is there any plainly evident proof of the economy of high pressure steam?

A. That high steam pressures tend to economy is plainly shown in ocean steamers. Twenty-five years ago they carried 16 lbs. pressure per square inch, and it took 5 to 6 lbs. of coal per hour to make a H. P. Now they carry 75 lbs. and get one H. P. per hour with 2½ to 3 lbs. of coal. The "Anthracite," carrying 350 to 500 lbs. pressure, crossed the Atlantic with only 1 lb. of coal per hour per H. P.

Q. Under what circumstances may small ports be more economical of steam than large ones?

A. In some cases (as, for instance, in locomotive engines with badly designed valves and valve gear) where small ports, by wire drawing the steam, practically effect the expansion of the steam before the lap of the valve cuts off the admission.

Q. How is it that some people find inside lap to save coal and others find it do just the opposite?

A. The effect of inside lap, or exhaust lap, as it is variously termed, is to diminish the exhaust passage, and after a certain portion of the stroke to completely close it. Thus we have two different effects upon economy.

The first is a tendency to produce back pressure if the engine be running very fast; and back pressure previous to compression is always a waste of steam. So that if one valve had too little travel, or too little arch in the valve, or too small an exhaust passage, cutting out inside lap might free the exhaust and thus increase the economy of steam.

The second effect of exhaust lap is to cushion the exhaust at a certain portion of the return stroke; and this cushion or compression, if in proper amount, has for its effect to save the waste steam contained in the clearance spaces, and which would otherwise reduce the grade of expansion and also chill down the steam entering from the steam-chest. The higher the speed (so long as the exhaust passages are large enough and there is enough exhaust lead) the greater the benefit of exhaust lap simply as giving a cushion to the reciprocating parts.

The greater the waste clearance space, the greater the benefit of exhaust lap up to that point, at which point the exhaust steam in the waste spaces is compressed up to steam-chest pressure. The higher the steam-chest pressure the greater the benefit of exhaust lap, because the greater the compression of exhaust steam in the waste clearance space.

Thus both may be right.

"Of two evils, the least is always to be chosen."

Q. What are the causes and effects of wire-drawing?

A. Unsteady wire-drawing has as its cause, insufficient size of ports and passages, and as its injurious effect, reduction of the mean effective pressure and of the ratio between this and the terminal pressure.

Q. Does wire-drawing necessarily wholly represent wasted energy?

A. Wire-drawing does not entirely waste energy, as while it lowers the initial pressure it causes slight super-heating; but it lessens the ratio between terminal and back pressures, and lessens the expansion ratio.

Q. What is the most advantageous adjustment of compression or cushion?

A. The most advantageous adjustment of compression or cushion, (from the point of view of steam economy, and entirely irrespective of the question of pounding) is when the quantity of steam cushioned just fills clearance spaces at initial pressure.

Q. Why will not compression save all the loss by clearance?

A. Compression cannot save all the loss by clearance, because the space through which the piston travels during compression is lost for the production of work, while friction continues.

Q. What is the amount of compression to give, for maximum steam economy?

A. For every point of cut-off and for every amount of back pressure, there is with each per cent of clearance, a point of compression which gives the best steam economy.

The following table by D. K. Clark, gives the best period of compression with 7% clearance, and for several back exhaust pressures and for several points of cut-off:

COMPRESSION OF STEAM IN THE CYLINDER; BEST PERIODS OF COMPRESSION.

Total back pressure, in parts of the total initial pressure at vacuum.	Cut off in percentages of stroke.						
	10	20	30	40	50	60	75
	Period of compression, in parts of stroke.						
Per cent.	per ct	per ct	per ct	per ct	per ct	per ct	per ct
2½	65	52	42½	35	29½	23½	16
5	57	47	39	32½	27	22½	15
10	45	40	34	29½	25	20½	14
15	37	33	29	25½	22½	18½	13½
20	—	27	26	23	21	17	13
25	—	23	22	21	18½	16	12½
30	—	—	19½	19	17½	15½	12
35	—	—	17½	16½	16	14½	11½

Q. What is the advantage of exhaust pre-release?

A. Exhaust pre-release enables the entire fall of pressure to take place towards the end of the stroke, effecting a saving in one part of the stroke and a loss in another. By a proper point of release the saving may be made to considerably exceed the loss.

Q. What is the point for exhaust release, at which the greatest saving is effected?

A. The greatest saving in work is effected by making the release at such a point that one-half the fall of pressure shall take place at the end of the forward stroke and one-half at the commencement of the return stroke.

Q. What is the effect of engine room temperature on cylinder condensation?

A. The lower the engine room temperature, the greater the amount of cylinder condensation.

Q. What are the advantages of a small engine over an underloaded large engine?

A. As the friction of the engine and transmission, does not greatly vary with the load; as the friction of a small engine is

less than that of a large one driving the same load, it is best to use a small engine at maximum economic capacity, than a large one, underloaded.

The following figures are given by Mr. Julius L. Hornig as from indicator cards:

COMPARISON TABLE OF USEFUL PERCENTAGE UP TO FULL CAPACITY:

Work divisions.	1	2	3	4	5	6	7	8	9	10
Per cent for large engine.	35	55	66	75	80	86	92	95	98	100
Per cent for small engine.	55	75	86	95	100					
Per ct. difference.	20	20	20	20	20					

Thus the small engine, with only half the capacity of the large one, shows a gain of 20% if used instead of the large engine, when the latter is worked at half its capacity or less.

Q. How does the exhaust waste fuel?

A. Every steam engineer should look upon the exhaust as one of the most active and competent of the seven thieves which are robbing his engine of heat and his

pocket of cash. The exhaust waste is by reason of the heat taken up by the exhaust from the interior surfaces of the cylinder walls, which thus, instead of remaining hot, become chilled and cause internal condensation during admission and expansion.

Q. Is lagging the exhaust pipe any use?

A. No.

Q. Is air space between lagging and cylinder, advisable?

A. Yes.

Q. Does the loss of a steam engine by friction vary with the load?

A. Not necessarily.

Q. What is the effect of a draft through the engine room?

A. To increase cylinder condensation.

Q. Which is the more economical method of lowering the capacity of an engine:—throttling, or cutting off earlier?

A. In general, earlier cut-off is the more economical of steam, because throttling causes wire-drawing and lowers initial pressure, while earlier cut-off only diminishes the mean effective pressure.

CAPACITY OR POWER.

Q. What is meant by the "capacity" or "power" of an engine?

A. The capacity or power of an engine is the amount of work it can do, irrespective of economy. Thus, a 22″ × 36″ engine running 110 revolutions (660′ piston speed) per minute, has, at 80 lbs. initial pressure, a capacity of about 397 H. P.; but it is more economical, that is, uses less steam per hour per H. P., when worked to from 225 to 342 H. P.

Q. How may the horse power of a steam engine be expressed?

A. The horse power of a steam engine may be expressed by Prof. Marks' formula:

$$(HP) = \frac{PLAN}{33{,}000};$$

(HP) being the indicated horse power; P, the mean steam pressure on piston, in lbs. per square inch; L, the stroke in feet; A, the piston area in square inches, and N, the number of strokes (or double the number of crank revolutions) per minute. The

author has a somewhat shorter formula:

$$(HP) = \frac{PAT}{33,000};$$

in which the term T, representing the piston travel in feet per minute, replaces the two terms L and N, in the preceding formula.

Q. What is a nominal horse power?

A. "Nominal horse power" does not mean anything in particular. It corresponds to "as long as a piece of string;" or "as big as a lump of chalk."

Q. What is an actual or real horse power?

A. An actual or real horse power is 550' lbs. per second, or 33,000 foot lbs. per minute, or 1,980,000 foot lbs. per hour.

Q. What is a "*force de cheval*" or "*cheval vapeur?*"

A. The "*force de cheval*" or "*cheval vapeur,*" is the French horse power. It is

			ft. lbs.
75 kilogrammeters	per	second	=542½
or 4,500	"	" minute	=32,549
270,000	"	" hour	=1,952,932

or about $\frac{1}{70}$ part less than the British horse power.

Q. How can you ascertain the gross horse power from the piston area, quickly?

A. Look in the following table, in the square corresponding to the mean effective pressure and piston speed of your engine, and you have a number, which, multiplied by the piston area, gives the gross horse power.

MULTIPLIERS FOR VARIOUS SPEEDS AND PRESSURES:

Lbs M. E. P.	Lineal piston speeds in feet per minute.				
	300	400	500	600	700
10	.090909	.121212	.151515	.181818	.212121
15	.136364	.181818	.241818	.272727	.318182
20	.181818	.242424	.303030	.363636	.424242
25	.227273	.303030	.378788	.454545	.530303
30	.272727	.363636	.454545	.545455	.636364
35	.318182	.424242	.530303	.636364	.742424
40	.363636	.484848	.606061	.727273	.848485
45	.409091	.545455	.681818	.828283	.954545
50	.454545	.606061	.757576	.909091	1.060606
55	.500000	.666667	.833333	1.000000	1.166667
60	.545455	.727273	.909091	1.090909	1.272727

Thus:—an engine 10″ bore, with 40 lbs. mean effective pressure and 400′ piston speed, has a gross horse power of 78.54× .484848=37.75.

Q. What is the "factor of horse power" of an engine?

Q. How can you ascertain the gross horse power from the piston area, quickly?

A. Look in the following table, in the square corresponding to the mean effective pressure and piston speed of your engine, and you have a number, which, multiplied by the piston area, gives the gross horse power.

MULTIPLIERS FOR VARIOUS SPEEDS AND PRESSURES:

Lbs M. E. P.	Lineal piston speeds in feet per minute.				
	300	400	500	600	700
10	.0909[illegible]	.[illegible]21212	.151515	.181818	.212121
15	.13636[illegible]	[illegible]1818	.241818	.272727	.318182
20	.181818	[illegible]2424	.303030	.363636	.424242
25	.22727[illegible]	[illegible]03030	.378788	.454545	.530303
30	.272727	[illegible]	.454545	.545455	.636364
35	.318182	[illegible]12	.530303	.636364	.742424
[illegible]	[illegible]	[illegible]	.606061	.727273	.848485
[illegible]	[illegible]	[illegible]	[illegible]1818	.828283	.954545
[illegible]	[illegible]	[illegible]	[illegible]7576	.909091	1.060606
[illegible]	[illegible]	[illegible]	[illegible]3	1.000000	1.166667
[illegible]	[illegible]	[illegible]	[illegible]91	1.090909	1.272727

[illegible] bore, with 40 lbs.
[illegible] and 400′ piston
[illegible] power of 78.54×

[illegible] of horse power"

A. The factor of horse power of an engine is the product of the area of the piston in square inches, by its speed in feet per minute, divided by 33,000. This figure, multiplied by the mean effective pressure, gives the gross horse power of the engine. Thus, a 14″ × 24″ engine, making 150 turns per minute, has a factor of horse power of

$$\frac{(14 \times 14 \times .7854) \times (2 \times 300)}{33,000} = 2.79$$

and with 40 lbs. mean effective pressure the gross horse power is $40 \times 2.79 = 111.7$.

I contributed the following some time ago to the *Millstone*, and reproduce it in order that my readers may construct similar tables for their own use:—

"We often see even engine builders take a long while ciphering up the horse power of their standard sizes of engines at various speeds and mean effective pressures, and wonder that they do not have a "factor of horse power" calculated for each engine at its rated rotatum speed, so that its horse power, with any given mean effective pressure, may be obtained in a moment by one simple multiplication. We give herewith

the factors of horse power of a series of engines that we get up for "direct driving," electric light machines, circular saw mills, &c. It will be noted that the rotation speeds are unusually light; but the engines are guaranteed to run cool and steady, and govern closely even under the excessive changes of load usual in such work.

The factor of horse power is from

"Grimshaw's Formula" $\frac{PAT}{33{,}000}$; being $\frac{AT}{33{,}000}$;

that is, the quotient of 33,000 into the piston area (in square inches) times the travel (in feet per minute).

Diameter.	Stroke, Inches.	Revolutions.	Piston Speed.	Area.	Factor Horse Power.	Horse Power at 50 lbs. Mean effective pressure.
3	3	1,200	600	7.07	.12855	6.427
4	4	900	600	12.57	.22855	11.427
5	5	750	625	19.64	.37196	18.598
6		750	625	28.27	.53571	26.785
6		650	650	28.27	.55683	27.841
8		525	700	50.27	1.06606	53.303

Q. What is the "horse power constant" of an engine?

A. The "horse power constant" of an engine is generally understood to mean the horse power developed for each revolution per minute and each pound per square inch mean effective pressure. Thus, a 12″ × 21″ "Hartford" engine, under an initial steam pressure of 80 gauge lbs., and cutting off at $\frac{1}{5}$ stroke, (say 35 lbs. mean effective pressure) would, at 170 revolutions per minute=595 piston speed per minute, have a horse power constant of .4139; and its horse power would be 170×35×.4139= 70.36.

Q. What is an easy rule for gross and net horse power of engines of 12″ bore?"

A. *In all engines of* 12″ *bore* the gross horse power equals the products of the mean effective pressure and the travel in feet per minute, divided by 291.8; and the net horse power is about equal to the mean effective pressure times thrice the lineal piston speed, with three figures pointed off from the right.

Thus:—12″ engine with 40 lbs. mean effective pressure, and 400′ piston speed;

then the net horse power$=40\times 1,200=$ 48.

A simple formula for horse power of engines is

$$\frac{P\,A\,T}{33,000}$$

P being the mean effective pressure in pounds per square inch, A the piston area in square inches, and T the piston travel in feet per minute.

Q. Is there any usual mean effective pressure, and piston speed, which makes it especially easy to calculate gross horse power from piston area?

A. Referring to the table,* it will be seen that the multiplier for 55 lbs. mean effective pressure, and 600′ piston speed, is 1; so that in any engine working at those rates (which are customary inactual practice) the gross horse power is the same as the square inches of piston area.

Q. Is it fair to rate the power of an engine by its ability to drive a certain machine, rated to take a given horse power?

* This table will be found on page 87.

A. No. The indicator, dynamometer, and the brake offer the only fair method of rating an engine. For instance, some descriptions of electric light machines sold to produce a given candle power with a stated number of horse power, consume two to ten times the stipulated motive force.

Q. What is a hyperbolic logarithm, and how can I find the logarithm or hyperbolic logarithm of any number?

A. You understand that a "power" of a number is that number multiplied by itself a certain number of times. Thus $2\times2=4$ is the *second* power of 2; $2\times2\times2=8$ is the *third* power of 2; $2\times2\times2\times2\times2=32$ is the *fifth* power of 2. Sometimes they are written thus:—2^2 for second power of 2; 2^3 for third power of 2; 2^5 for fifth power of 2, and so on. The little figure showing how many times the number is taken as a factor is called the "exponent," or "shower."

Now note that if you multiply 2^2 or the *second* power of 2, by 2^3 or the *third* power of 2, you get 2^5 or the *fifth* power of 2; that is, in multiplying powers of a

number together you *add* the exponents.

Conversely, in dividing one power of a number by another power of a number you *subtract* the exponent of the divisor from the exponent of the dividend. Thus, 2^7 divided by 2^3 gives 2^4; that is, $2\times2\times2\times2\times2\times2\times2$ (or 128) divided by $2\times2\times2$ (or 8) gives $2\times2\times2\times2$ (or 16).

The "reverse" of a power is a "root." That is, if 25 is the second power of 5, then 5 is the second root of 25; if 81 is the fourth power of 3, then 3 is the fourth root of 81.

A root may be expressed by writing its exponent as the denominator of a fraction up at the right hand top corner of the number; thus $81^{1/4}$ means the fourth root, or reverse of the fourth power, of 81.

Now suppose we want to take the third power of the fourth root of 3; or the fourth root of the third power of 3. We can express that $3^{3/4}$; that is a fractional exponent. Let us work out the fourth root of the third power of 3, so we can see how much it is: $3^3=3\times3\times3=27$.

$3^{3/4}=27^{1/4}$, got by the following operation:

```
    27.00′00(5.1961 is the second root
    25.                          of 27.
    ——
101)200
    101
    ———
1029)9900
     9261
     ————
10386)63900
      62316
      —————
103921)158400
       103921
       ——————
```

```
   5.19′61)2.2794 is the second root of
   4.              5.1961 or the fourth
   —               root of 27, or the
42)119             fourth root of the
   84              third power of 3, or
   ——              is 3¾, or is ⁴√3³, as
447)3561           it may also be writ-
    3129           ten.
    ————
4549)43200
     40941
     —————
45584)225900
      182336
      ——————
```

5.19′61)2.2794 is the *second* root of 5.1961 or the fourth root of 27, or the fourth root of the third power of 3, or is $3^{3/4}$, or is $\sqrt[4]{3^3}$, as it may also be written.

Now suppose we write $\frac{3}{4}$ decimally as 0.75, and put it $3^{0.75}$; then we have a decimal exponent for the 3.

You can easily see that every number must be some root of some power of some other number; that is, must be an exponent (generally a fractional exponent) of some other number taken as a "base."

Thus, if we take 10 as a "base," then 2 is equal to the hundred thousandth root of the 30,103d power of 10, and may be written $10.^{0.30103}$; 3 is equal to $10.^{0.47712}$; and 6, or 3×2, equals $10.^{0.77815}$. If you will add 30,103 and 47,712 you will get 77,815.

Now there are tables prepared which show what power and root of 10, every number is; so that if we want to multiply any long number by any other long number, we turn to the table, get these "exponents," add them, and then look in the table and see what number has for its exponent of 10, the sum of these exponents, and that number is the product of the two numbers.

These exponents of 10 are called common logarithms; there are plenty of tables of

them in mathematical works, and they save an immense deal of labor, besides ensuring greater accuracy than where long calculations are gone through with.

Here we "do an example" in multiplication by common logarithms:

Multiply	1.23456, of which the com. log. is....	0.090067
By	7.89883 " " " " " "....	0.897563
	The sum of the logarithms is..........	0.987634

of which the "natural number" is 9.919177.

Now we'll try an example of *division* by logarithms:

Divide	187.8562, of which the com log. is.....	2.272041
By	34.5999, of which the com. log. is....	1.557944
	The difference of the logarithms is.....	0.734097

of which the natural number is 5.421225.

Now to take any power of a number: Suppose we want the fifth power of

184.9263, of which the common logarithm is..	2.266999
Then we multiply that logarithm by.......	5
The product is......................	11.334995

having for its "natural number" 216,269,-203,980.09+, which is the fifth power of 184.9263, as you can prove by covering a few square yards of paper with figures.

To take any root of a number: Suppose we want the seventh root of 9,999.999, of

which the common logarithm is 3.999999. Dividing 3.999999 by 7 we get 0.571428, which is the common logarithm of 3.7276. Then 3.7276 is the seventh root of 9,999.999.

The "hyperbolic logarithm," or "Napierian logarithm," of any number is the "common logarithm" multiplied by 2.3025851; there are tables of hyperbolic logarithms in most of the important works which treat of steam, and they are just as essential to the steam engineer as the common logarithms are to the scientific man who works in other lines. They *shorten labor.*

Thus, suppose an actual expansion rate of two in a cylinder 2′ long, and one square foot area of piston. Then at point of cut off we have one cubic foot of steam. Say it is at 100 lbs. pressure above vacuum, and the expansion is perfect. Then at full stroke we have 2 cubic feet of steam at 50 lbs. At $\frac{3}{4}$ stroke we have 1.5 cubic feet at $66\frac{2}{3}$ lbs. At $\frac{5}{8}$ stroke we have 1.25 cubic feet at 80 lbs. At $\frac{7}{8}$ we have 1.75 cubic feet at 57.14 lbs.; and so on.

To *roughly approximate* the average pressure in the cylinder during the time of expansion, we may thus proceed:

At $\frac{5}{8}$ stroke it is.................	80	lbs.
" $\frac{3}{4}$ " "	66.67	"
" $\frac{7}{8}$ " "	57.14	"
" $\frac{8}{8}$ or full stroke it is..........	50.	"
	4)253.81 lbs.	

Rough approximate average....63.45 lbs.

But suppose we count it at the $\frac{1}{16}$ths instead of at the $\frac{1}{8}$ths, it ciphers up

At $\frac{9}{16}$ stroke there are $\frac{9}{8}$=1.125 cubic feet; pressure $\frac{8}{9}$ of 100=	88.88lb
At $\frac{5}{8}$ stroke there are $\frac{5}{4}$=1.25 cubic feet; pressure $\frac{4}{5}$ of 100=	80. lb
At $\frac{11}{16}$ stroke there are $\frac{11}{8}$=1.375 cubic feet; pressure $\frac{8}{11}$ of 100=	72.73lb
At $\frac{3}{4}$ stroke there are $\frac{3}{2}$=1.5 cubic feet; pressure $\frac{2}{3}$ of 100=	66.67lb
At $\frac{13}{16}$ stroke there are $\frac{13}{8}$=1.625 cubic feet; pressure $\frac{8}{13}$ of 100=	61.54lb
At $\frac{7}{8}$ stroke there are $\frac{7}{4}$=1.75 cubic feet; pressure $\frac{4}{7}$ of 100=	57.14lb
At $\frac{15}{16}$ stroke there are $\frac{15}{8}$=1.875 cubic feet; pressure $\frac{8}{15}$ of 100=	53.33lb
At $\frac{16}{16}$ or full stroke there are $\frac{16}{8}$=2 cubic feet; pressure $\frac{8}{16}$ of 100=	50. lb
	8)530.29lb

Second approximate average, 66.28lb

Now the table of hyperbolic logarithms says that the hyperbolic logarithm of 2

(the expansion rate) is .6931472; so that without all the foregoing figuring, which after all only gives a rough guess at the result, we may get the exact average pressure as 69.31472 lbs.

THE CONDENSER

Q. What is a vacuum?

A. A vacuum is an absolutely empty space; although as we ordinarily understand it, it means a closed space from which a part of the contents has been abstracted, and which contains nothing but air or other gaseous fluid (or a mixture thereof), at a less tension than that of the atmosphere.

Q. What is the pressure in a vacuum?

A. The pressure or tension in a perfect vacuum is nothing, outwards; but there is an unbalanced pressure of nearly 15 lbs. per square inch (at the sea level) from without, towards its center, tending to fill it with whatever fluid surrounds it.

Q. What is it which exerts this unbalanced pressure of nearly 15 lbs. per square inch from without inwards, on the vessel containing a vacuum?

A. The weight of the atmosphere. A

column of air of one square inch at the base, and running from the sea-level to the upper limits of the atmosphere, weighs 14.7 lbs.

Q. With what velocity does air rush into a perfect vacuum?

A. 1,338 feet per second.

Q. What is the use of the so called air pump, in a condensing engine?

A. The "air pump" removes the water from the condenser to prevent the air which it contains, accumulating and still further destroying the vacuum.

Q. Could a condensing engine be worked with a steam pressure less than that of the atmosphere?

A. Yes, if once started.

Q. What would be the objection against running a condensing engine with steam at less than 14.7 lbs. per square inch in the boiler?

A. The boiler could not be blown out; and the gauge cocks could not be tried.

Q. When and by whom was the condenser first applied to decrease exhaust pressure?

A. By James Watt in 1765.

Q. How many principal types of condensers are there; and what are they?

A. Four:—the surface condenser of marine vessels:* the air pump jet condenser;† the ordinary siphon condenser, and the exhaust steam induction condenser.

Q. What is a "siphon condenser?" ‡

A. A "siphon condenser" is one in which the exhaust steam escapes through a nozzle of special shape, surrounded by another through which flows with a fall of 34′ an annular current of water which condenses the exhaust steam.

Q. What is an exhaust steam induction condenser?

A. A modification of the siphon condenser in which is utilized the velocity of the exhaust steam. It provides for its own water supply without the aid of a pump.

Q. How is the exhaust steam induction condenser started?

A. By a small jet of live steam, which is shut off after the machine is started.

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* See Steam Engine Catechism, page 74.
† See Steam Engine Catechism, page 73.
‡ See Steam Engine Catechism, page 78.

Q. What is essential in a so called siphon condenser?

A. That the condensing water shall properly fill the cone or jet, but without overcrowding. If the cone be too large, too much water will be needed.

STEAM ENGINE RATING.

Q. What are the principal elements to be considered in choosing or comparing steam engines?

A. The principal elements to be considered in choosing or comparing steam engines, are their duty or economy, simplicity, regulation, and compactness.

Q. What are the relative values of these four principal elements?

A. This varies with the use to which the engine is to be put, and the conditions under which it is to be run. Opinions of engineers and engine buyers differ widely. The following opinions from eminent authorities will prove of interest in this connection:

The Harlan & Hollingsworth Co., Wilmington, Del., builders of marine engines, while saying that the conditions indicated by such figures are necessarily modified in

different engines, estimate as follows: Economy, 40; durability, 20; simplicity, 20; regulation, 25; and compactness, 5.

Mr. J. W. See, Hamilton, Ohio, does not think it proper at all to divide a possible 100 points among the features stated on any scale, as some other features entitled to recognition in the hundred might prove to have been omitted; and he doubts any one built to give each feature its proper percentage. He would say that a percentage based upon a possible hundred of each feature now thought of or turning up in the future would be nearer right. In six out of every ten engines Mr. See thinks that regulation would be a matter of no import whatever, and in some economy would be nothing.

The Norwalk Iron Works, South Norwalk, Conn., by E. Hill, Treasurer, say that regarding the rating of engines they would judge that no fixed rule could be adopted. In their opinion for engines intended for use in large cities and vicinity, economy would stand first, and in crowded buildings compactness would always be an important feature. Simplicity there is of not much

Q. What is
condenser?
A. That the
only fill the
exceeding. If
much water

Q. What
considered in
engines?
A. The p
ered in cho
gines, are th
regulation,
Q. What
four princip
A. This
engine is

engines, estimate as follows: y, 40; durability, 20; simplicity, lation, 25; and compactness, 5.

W. See, Hamilton, Ohio, does not proper at all to divide a possible among the features stated on any some other features entitled to on in the hundred might prove to omitted; and he doubts any one give each feature its proper per- He would say that a percentage a possible hundred of each fea- thought of or turning up in the ould be nearer right. In six out en engines Mr. See thinks that would be a matter of no import e, and in some economy wou

Works, S

l, Treasur

engine

could

gine

account, especially on a large engine, as skilled help is easily obtained. Durability would stand low in the scale, as repairs are easily made, and proprietors are progressive and ready to introduce improved machines as they appear.

For flour mill engines and many similar industries, regulation is the first point to be considered, as the quality of the entire product depends upon that. The other points would then be affected by other circumstances of the case. For a coal mine durability stands pre-eminent, and next, simplicity, and if the power is transferred to a long distance through wire ropes, regulation should be first-class.

Mine engines on barren mountain peaks, difficult of access, demand durability and economy.

The Atlas Engine Works, Indianapolis, Ind., say concerning the value of the various features that make up the performance of an engine, they suppose that they would vary with different classes of engineers for different services. For stationary engines for ordinary land purposes their ideas would be to divide the 100 points thus:

Economy, 40; durability, 25; simplicity, 10; regulation, 20; compactness, 5.

Prof. de Volson Wood says, in this connection, that special conditions will greatly affect in particular elements. For average cases he thinks that adjustability will be of but little value, and in the hands of an ignorant runner it may be a damage and should (for him) be marked 0; a few intelligent runners might prize it highly. Similarly, in some cases, a want of compactness would render the engine nearly worthless for that particular case.

Such being the case, the judgment of different engineers, Prof. Wood thinks, will differ greatly, but if he were to rate the elements with a definite number, instead of giving a range, he would mark them about thus: Economy, 30; durability, 40; simplicity, 15; regulation, 5; compactness, 10. Still, he is not certain how much may be included in regulation in this schedule—such as automatic cut off, or mechanical means for taking up wear.

Messrs. O. A. Pray & Co., Minneapolis, Minn., rate economy at 50; durability, 35; simplicity, 5; regulation, 5; compaatness, 5.

The Fitchburg Steam Engine Company would divide about as follows: Economy, 50%; durability, 16; regulation, 16; simplicity, 13; compactness, 5. They think the question has many side issues upon which each depends; hence no fixed rule can be adopted for all conditions. Economy is dependent to a great degree upon close regulation under all variations of load and pressure, avoiding spasmodic use of steam; and also upon simplicity, for although, under critical tests, any engine may control well and all parts be in perfect adjustment, yet the lack of simplicity allows speedy and perhaps disastrous misadjustment, so that the difference between the critical and the average performance for five years may be largely in favor of a simple engine with a much lower first duty.

Mr. F. F. Hemenway says that in the little thought he has as yet been able to give the subject, he must confess that the difficulties, mainly, have presented themselves to him. For instance, economy with coal at $10, and with sawdust to throw away, would be two different things; in like manner regulation may be very important

in one instance and of not much in another. Mr. Hemenway says that if he were to do the same thing he would, if he wanted to make it generally acceptable to engineers, make his own figures to the best of his knowledge and then ask the opinions of all he thought best, requesting them to alter the figures and give reason for so doing.

DESIGN, CONSTRUCTION AND ERECTION.

[See also both in other parts of this Supplement, and in the Steam Engine Catechism, under the heads of the various parts of the engine.]

Q. What is a rotative engine?

A. One in which the reciprocation of the piston in a straight line causes the crank and shaft to turn.

Q. What is a rotatory engine?

A. A rotatory engine is one in which there is no reciprocating (or "to and fro") piston, but in which the action of the steam causes the shaft to turn, without the intervention of intermediate mechanism.

Q. What type of valve causes the most wire-drawing?

A. The slide valve causes the most wire-drawing.

Q. What are the advantages of the inverted vertical type of stationary engine?

A. Lower frictional resistance, smaller amount of floor space, higher speed, and less foundation needed, than for a horizontal engine. Besides, it can be erected in a more contracted position difficult of access.

Q. What is meant by a "Corliss" type of engine?

A. In the United States, an engine with the rotative Corliss valve; in Europe, any engine with cut-off regulated by the governor.

Q. What is the advantage of "conical" or tapered valves?

A. "Conical" valves have the advantage that they may have wear taken up by end adjustment.

Q. What is the objection to conical valves?

A. "Conical" (or tapered) valves, unless hung on centres or on hardened trunnions, wear faster at the large end, and leak there, while they cannot be "set up" without pinching the small end.

Q. What are the disadvantages of two-ported engines?

A. One disadvantage of having but one port for each admission and exhaust is that the cool expanded steam cools the metal of the port and lowers the temperature of the next admission.

Q. How are cylinders and steam-chests generally protected from external radiation?

A. By wooden "lagging" (made of strips held on by metal bands or covers) or by coarse hair felt with metal casing.

Q. Is cork a good lagging for steam pipes?

A. Cork is a good lagging for steam pipes if kept from charring by the interposition of asbestos paper or asbestos blanket.

Q. What is the disadvantage of having the eccentric keyed on the shaft?

A. Adjustment is very difficult.

Q. What is the disadvantage of having the eccentric fastened by set screws?

A. They are liable to slip.

Q. How is the amount of lead determined?

A. The proper amount of lead must be

determined by experiment with each engine. Twin engines run at differing speed rates or under differing pressures or with steam of differing dryness, require differing leads.

Q. What is "setting" slide valves?

A. Fastening the eccentrics in proper position on the shaft (or axle) and adjusting the length of eccentric rods and valve stems.

Q. Where should steam lead be greatest, and why?

A. Steam lead should be greatest on high speed engines and on beam engines, because these are most likely to thump, by reason of their greater momentum.

Q. Where should exhaust lead be greatest?

A. Exhaust lead should be greatest with high speed engines and late cut-off.

Q. Which is generally greater, exhaust lead or steam lead?

A. Exhaust lead is generally greater than steam lead, because exhaust lead has to release all the steam in the cylinder, while steam lead is only to fill the clearance with live steam.

Q. How should a slide valve be ground?

A. In grinding a slide valve in which there are hollow places, it is better to use water with the emery or sand than oil; and kerosene is better than any other oil for this purpose.

Q. What is necessary before either release or admission can take place?

A. The valve movement must exceed the exhaust lap before the exhaust is open; and must exceed the steam lap before admission can commence.

Q. How may the port opening for any amount of valve movement, be calculated?

A. To get the port opening for any amount of valve-throw, subtract from that throw, the steam lap or the exhaust lap, whichever governs the case.

Q. What should be the rule for determining steam port area?

A. The steam port area of an engine must be sufficient to give the steam ample inlet without wire-drawing, and exit without causing excessive pressure. The steam passages must not be so l rge as to cause excessive clearance. For a piston speed of 600′ per minute, the area of each port

ought to be about $\frac{1}{10}$ the piston area, with ordinary steam; $\frac{1}{12}$ will answer where the steam is very dry. For other piston speeds the area should be proportional.

Q. What easy, approximate rule can you give for area of ports of "square" engines?

A. For "square" engines running 600′ per minute, with 55 lbs. mean effective pressure, two square inches of port area per horse power, or $\frac{1}{100}$ square inch for every cubic inch of cylinder capacity, is about right; and other speeds and mean effective pressures, proportionally.

Q. What is the objection to large port area?

A. The objection to large port area is that it causes excessive clearance.

Q. What is the objection to small port area?

A. The objection to small port area is that it causes excessive friction.

Q. How is the steam port width determined?

A. The steam port width is found by dividing the necessary area by the greatest practicable length.

Q. What are the objections to a very narrow exhaust port?

A. A very narrow exhaust port will choke the exhaust at the end of the valve stroke.

Q. What are the objections to a very wide exhaust?

A. A very wide exhaust necessitates an unnecessarily long slide valve, and excessive friction.

Q. What is gained by having the steam-chest under the cylinder?

A. The special use of having the steam-chest under the cylinder, is to free the cylinder readily of water of condensation, which would otherwise endanger the cylinder heads.

Q, What are the evils of too small crank pin diameter?

A. Too slender a crank pin is liable to bend or break.

Q. What are the objections to too large a crank pin?

A. Too large a crank pin heats; one of the greatest troubles about a steam engine.

Q. What objections to a very long crank pin?

A. A very long crank pin is liable to bend.

Q. What is the disadvantage of a very short crank pin?

A. A very short crank pin has not enough bearing surface.

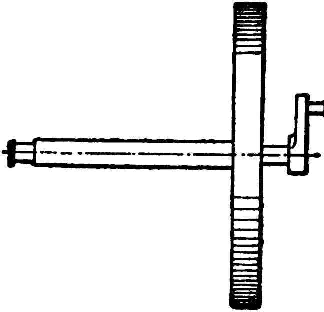


Fig. 9.

Q. What is the best proportion between fly-wheel weight and the work on the piston?

A. Mr. N. J. Raffard has examined into the best proportions between fly-wheel weight and the work on the piston; also the best place for the fly-wheel on the main shaft, as regards friction; and the superiority, from this point of view, of the horizontal over the vertical engine.

Supposing an engine carrying full steam, and with connecting rod of infinite length, and without fly-wheel, and such equal work as would be put on it by a completely universal screw propeller. For each position of the crank, the work carried by the connecting rod being constant in amount and direction, the pressure of the shaft against the brasses would be constant, and the friction

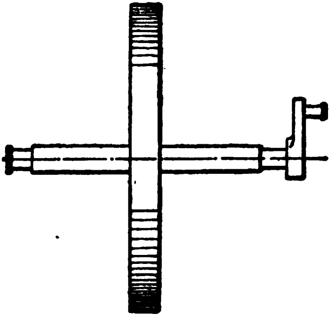


Fig. 10.

the same throughout the whole revolution, no matter what the inclination of the cylinder.

Adding a fly-wheel, as in Fig. 9, near the crank, its weight comes into play. In the vertical engine this fly-wheel weight helps

or hinders the piston, according to the direction of motion; but this helping or hindering, in the case of the vertical engine, balance. Supposing the same engine arranged horizontally; there will be 40% more fly-wheel friction.

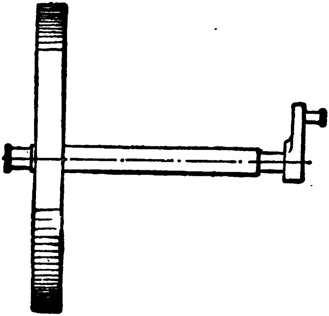


Fig. 11.

Placing the fly-wheel in the middle of the shaft as in Fig. 10, then in the vertical machine only fifty per cent. of the friction due to the fly-wheel weight will be neutralized; but in the horizontal engine, only 30%.

Putting the fly-wheel at the other end of the shaft (Fig. 11) the fly-wheel friction will be twice as much in the vertical engine as

in the same engine with fly-wheel next the crank.

Mr. Raffard applied these principles to the Woolf compound engine of the following dimensions, &c.:

Stroke,	............0.90	metres=	35.433	in.
Diam. L. P. cyl.	....0.40	"	15.748	"
" H. P. "	....0.18	"	7.087	"
" Shaft	........0.155	"	6.103	"
" Flywheel	.....4.50	"	13.12	ft.

Con. Rod, 5 cranks long.

Pressure, 5 atmospheres=say 73.5 lbs.

Cut-off at $\frac{1}{8}$ stroke.

Rev. per min., 40 to 45.

Theoretical ind. H. P., 60 H. P.

Weight of flywheel, (see table.)

The table gives in kilogrammeters* the friction in the bearings, coming from the combined efforts of the pressure on the connecting rod, and the weight of the fly-wheel, under various conditions of fly-wheel position relative to the crank; and neglecting that of the shaft itself; though taking into account the obliquity of the connecting rod:

*1 kilogrammeter equals .672 lb. per ft. or 2.016 per yard.

Fly wheel position on shaft.	No. 1, Vertical Engine, Fly wheel weighing 2400 kilog† — Friction in kilog§	No. 1, Vertical Engine, Fly wheel weighing 4800 kilog‡ — Friction in kilog	No. 2, Horizontal Engine. Fly wheel weighing 2400 kilog — Friction in kilog	No. 2, Horizontal Engine. Fly wheel weighing 4800 kilog — Friction in kilog
By the crank....	105.6	140.	117.5	162.2
Middle of shaft.....	138.4	172.2	139.6	184.2
Opposite crank.....	172.5	239.2	172.5	239.2

This table shows how much the friction can vary according to the disposition of the parts of the machine; and that the variation can run as high as 99 kilogrammeters per turn in an engine of about 30 H. P., even when the stroke is long as compared with the cylinder diameter.

This variation is considerably increased with the work in the pistons; that is to say, when with the same nominal power, the

† 4,891 lbs. av. ‡ 9,782 lbs. av.

§ Kilogram equals 2.2046 lbs. avoirdupois; 1 lb. avoirdupois equals 0.45359 kilograms.

stroke is diminished and the fly-wheel weight increased.

The Woolf "pendulum" engine with fly-

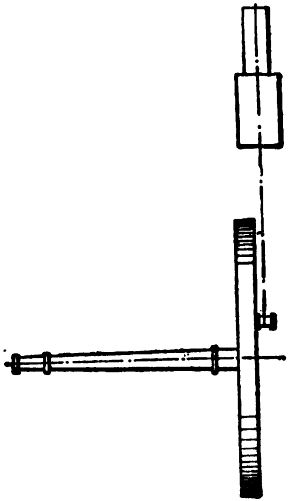


Fig 12.

wheel near the connecting rods (see Figs. 12, 13, 14,) and vertical cylinders, permits of reducing or even completely neutralizing fly-weeel friction.

Q. What is the best place for the fly-wheel on the main shaft, as regarcs friction?

A. Next the crank. See account of

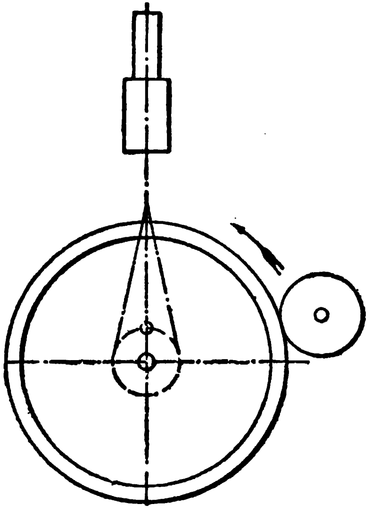


Fig 13.

Raffard's experiments, page 114.

Q. Which has the least fly-wheel friction, a horizontal or the vertical engine, other things being equal?

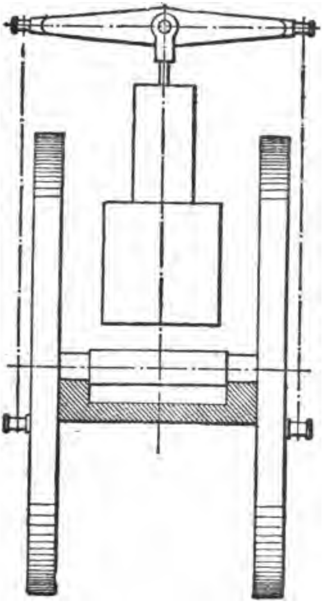


Fig. 14.

A. The vertical. See account of Raffard's experiments, page 114.

CARE AND USE.

Q. When and how should the valves be finally set?

A. The valves should be finally set by reference to indicator diagrams, and while they are hot.

Q. What is the best way to keep piston valves tight?

A. The best way is to bore the valve chamber accurately, make a good plug fit, and when leaks commence, rebore and make a new valve.

Q. How can slide valves be cured of wearing curved?

A. Slive valves that wear curved can generally be prevented from that sort of thing, after valve and seat have been planed up, by using a "back rod" as a guide?

Q, How may cutting of cylinders and excessive friction be lessened?

A. Dry plumbago will improve the running and decrease the cutting of steam engine cylinders.

Q. How may wire drawing be lessened?

A. Wire drawing may be lessened by the use of double eccentrics and "gridiron"

slide valves, and by increasing the area of steam ports and passages.

Q. How may wire drawing be avoided?

A. Wire drawing may be avoided by using the Corliss types of valve gear, where the admission valves are suddenly opened and closed.

Q. What may be said of the practice of having the parts of an engine "about true enough" rather than absolutely true?

A. The only place for a piston rod is in the exact centre of the piston and of the cylinder; the only position of the crank shaft is absolutely at right angles with the cylinder axis, and the crank pin should be absolutely in line with the shaft.

Q. How should the guides be lined up?

A. Guides on steam engines should be lined up by the bore of the cylinder and not by the counter bore.

Q. Which wears an engine out most, for a given piston speed, frequent reciprocation or fast rotation?

A. An engine wears out by the reciprocation of the parts rather than by the rotation of the wheel and shaft.

Q. What is "lost motion?"

A. Lost motion is lcoseness of parts which are supposed to prevent fixed relative positions. Thus, if the connecting rod have lost motion at the cross head pin the piston starts and "takes up" that looseness before it moves the connecting rod.

Q. What often causes jarring at the time of cut off?

A. Having an overhead supply pipe with a right angle in its horizontal portions.

Q. How may the pound in an engine be located ?

A. The pound in an engine may be located by placing one end of a piece of $\frac{1}{4}''$ wire, 8″ long, in the teeth and applying the other to each end of the crank shaft bearings, cylinder, etc. Where there is most shock there will be the pound.

Q. What is the probable cause of pounding when the crank is on the dead centre?

A. Pounding at the end of the stroke when the crank is on the dead centre may come from a ridge in the cylinder, or at the guide bar end, or from the connecting rod brasses not being keyed up tight enough.

Q. What is the probable cause of pounding at half throw?

A. Pounding when the crank is at half throw will not be caused by any of the above, but will be apt to be occasioned by the crank pin not being parallel to the crank shaft, or from one or the other of the crank shaft journals being low.

Q. Where do horizontal piston rods wear most?

A. Horizontal rods wear most in the middle; the ends wearing elliptical ("oval" so called); the back end wearing most on the bottom and the crank end most on the top (where the engine throws under this is reversed.)

Q. What are the requisites of a piston rod packing?

A. A piston rod packing must allow the rod to move freely up, down or sidewise with little friction, with little wear of the rod or of the packing; and must make a steam tight joint under the highest pressure that there will be found in the cylinder.

Q. What defects are we likely to find in piston rods that are out of true?

A. Pistons are apt to be out of centre, to

be out of round and crooked; either one or the other, and sometimes all three; in addition to which they are apt to be smallest in the middle, and the back end will be worn most upon the bottom and the crank end most upon the top.

Q. What are the principal causes of piston rod packing and valve stem packing being in bad condition?

A. Among the principal causes of rod packings being in bad condition are excessive speed, and steam pressure; lack of alignment; crooked or bad rod; too short stuffing boxes; packing of inferior kind or quality, or badly put in, or insufficiently cared for; and in the case of the piston rod packing, leaky piston or too little clearance.

Q. What are the effects of high pressure on many rod packings?

A. High pressure, generally accompanied by high speed, and always by high temperature, causes baking and abrasion of many packings, especially fibrous vegetable ones.

Q. What does lack of alignment do to rod packings?

A. Lack of alignment puts excess of pressure on one side of the packing and tends to wear it out rapidly at that side.

Q. How do crooked rods affect packings?

A. Crooked rods put sudden pressures first on one side and then on another, of the packings, and unless they are adjustable, tend to tear them out.

Q. What precaution should be taken in the use of fibrous piston rod packing?

A. If fibrous material be used, care should be taken lest it contain grit, which will scratch the piston lengthwise and cause channels for the escape of steam, besides making more friction.

Q. What rule is there for thickness of coil packing?

A. Coil packing should be in thickness, half the difference between the diameters of rod and box. Thus a 3″ rod in a 4½″ box requires ¾″ packing.

Q. What is the effect of cutting the rod packing rings too long?

A. If the rod packing rings are too long they will not hug the rod, and the packing will not be steam tight.

Q. What is caused by cutting the rod packing rings too short?

A. If the rod packing rings are cut too short they will not meet, and leakage will result.

Q. What is the proper procedure in packing a stuffing box?

A. In packing a stuffing box with coil packing, put in as much packing as will just allow the gland to enter; screw up hard; then slacken.

Q. What should be done if the eccentric straps heat?

A. If the eccentric straps heat, the bolts should be slackened a very little; and when the engine stops the straps should be removed and scraped (not filed) to a true and smooth surface.

Q. What is the best cure and prevention of heated bearings?

A. Good dry "air floated" plumbago (black lead; graphite) *free from grit*, will cool any hot bearing that anything else will, and will prevent heating, better than anything else which we have ever seen.

Q. How can bright work best be protected?

A. Bright work may be much better protected by tallow and lime than by white lead and tallow. In any case there should be an excess of tallow, so that the covering may be readily wiped off with waste. I have used the German *putz-pomade* for brass and nickel plate, with great satisfaction.

Q. How can bright work be freed from rust?

A. Crocus cloth properly used, with a soft pine stick, removes rust from bright work before it has time to get deep into the material, and should be kept in every engine room. It must be remembered that *rust breeds rust.*

Q. How should tight nuts be moved?

A. By first loosening them with coal oil, if rusted.

Q. How often should indicator cards be taken?

A. Indicator cards should be taken daily from every engine of twenty-five H. P., or over. They *pay* in time, money, freedom from annoyance, &c.

Q. What important memoranda may be made concerning the care and use of the indicator?

A. Test your indicator by some standard gauge before taking any cards. Never connect the indicator to the blow-off hole in the cylinder. Indicate both ends of the cylinder, and the steam chest as well.

Q. What is the advantage of a small steam chest?

A. A small steam chest gives the throttle better control of the engine than a large one, because if there is a large one the steam therein may act by expansion between the point of complete throttling and that at which the lap of the valve causes cut-off. A small steam chest has also to recommend it, that it decreases loss of temperature and pressure from condensation.

Q. What may be said of the carrying powers of steam pipes of different diameters?

A. Doubling the diameter gives *six* times the carrying power for long distances, as proved by the experience of the New York Steam Heating Co.

Q. What should be the diameter of the exhaust pipe?

A. The exhaust pipe should be double the area of the steam pipe.

Q. How may the exhaust be deadened?

A. The exhaust of an engine may be deadened by running it through a large box filled with perforated heads, or with ordinary pebbles all of one size. If the box is large there will be no back pressure made. Another way is to utilize the exhaust to heat the feed water.

Q. How are slide valves and seats finished?

A. By planing, then filing, and scraping so that every part of the valve face touches the seat, in all positions of the valve.

Q. What objection is there to cast cranks?

A. Cast cranks will not permit of shrinking the crank pin in the eye, without danger of cracking.

Q. Are half round brasses in connecting rod ends, good things?

A. Half round brasses in connecting rod ends must have been designed by old Clootie. Avoid their use.

Q. Which is the greater cause of crank pins heating; increase of speed or increase of pressure? and why?

A. Within certain limits, an increase in

speed heats the crank pin more than a proportionate increase in pressure, because it expels the lubricant more rapidly.

A. What is the best way to put in a crank pin?

A. The best way to put in a crank pin is to taper it a little and drive it in slowly by hydraulic pressure.

Q. How may the shock of a sudden admission of high pressure steam be neutralized?

A. By making the piston and cross head very heavy so that the shock will be absorbed by reason of the inertia of the then motionless or nearly motionless mass of metal.

Q. Is the velocity of the crank pin and fly wheel constant at all parts of the stroke?

A. It should be; and is, practically.

Q. Is the velocity of the piston the same at all parts of the stroke?

A. No.

Q. Is the variation of the velocity of the piston regular or irregular?

A. Regular.

Q. When is the velocity of the piston the least?

A. At stroke ends, when it is absolutely *nothing*.

Q. When is the velocity of the piston the greatest?

A. When it is a little ahead of mid-stoke on the forward stroke of a stationary engine, and a little behind mid-stroke on the return stroke.

Q. What is the maximum speed of the piston?

A. The maximum piston speed is about $\frac{11}{7}$* of its mean speed.

MISCELLANEOUS.

Q. What are the advantages of steel piston rods?

A. Steel piston rods have to recommend them—greater smoothness than wrought iron rods and greater stiffness for a given diameter; or for a given stiffness—less weight and less reduction of the effective area of the piston on the crank side.

Q. How is the piston rod fastened to the piston head?

A. Its end is tapered and passes through

*Accurately $\frac{3.1416}{2} = 1.5708$.

a tapered hole in the piston head. The taper prevents its being pushed through too far, (even if there be no shoulder, as there generally is) and a key or a nut prevents its being pulled out.

Q. How is the piston head of a large engine generally constructed?

A. Of two cast iron (or bronze) pieces; one a "spider," to which the rod is fastened; the other a "follower plate" bolted to the spider by "follower bolts."

Q. How is the piston made to work steam tight in the cylinder bore?

A. By "packing" of one kind or another, in suitable circumferential grooves. Sometimes hemp strands are employed. Most usually there are cast iron split rings set breaking joint and spring into the grooves in the spider. These rings are kept out by either their own elasticity, or the steam pressure in the cylinder, or springs within the spider.

Q. How are the piston rod and the valve stem generally made to work steam tight in the cylinder head and valve chest end, respectively?

A. By passing them through a stuffing

box, consisting of a cylindrical chamber surrounding the rod or stem, so as to leave an annular space, which is filled with packing material, (vegetable or metal) and the pressure of which against the rod is adjusted by a "gland," fitting outside the rod and inside the stuffing box, the depth to which it penetrates, being regulated either by bolts or by the stuffing box and the gland being threaded.

Q. What should be required of rod packing?

A. Rod packing should accommodate itself without undue wear of rod or packing, to crooked, tapering, and fluted rods.

Q. How is the piston rod attached to the cross head?

A. By its end being tapered and inserted into a tapering hole in the cross head, and held in by a key.

Q. How is the connecting rod attached to the cross head?

A. To a pin, generally called a *wrist pin*, and cast in one piece with the cross head, each end bearing in one side of the "fork" of the cross head. (In the Straight Line engine the wrist pin is fast in the

connecting rod end and turns in bearings, at each end, in the cross-head fork.)

Q. Which guide wears the most, in a horizontal engine; and why?

A. In a horizontal engine which "runs over," the bottom guide wears the most, because on the out stroke the cross-head end of the connecting rod bears down, the connecting rod being in compression, and the crank pin above the center line; and on the return stroke the connecting rod is in tension and the crank pin below the center line, so that the cross-head end of the connecting rod is then pulled down.

If the engine "runs under" as in a locomotive engine when going ahead, the top guide gets the most wear.

Q. What may be said of the width of piston rings and piston-valve rings?

A. Piston rings and piston-valve rings should be wider than the ports which they pass, to prevent their catching in the latter.

Q. What are the ways of cutting piston packing rings?

A. The usual way is by a straight diagonal cut but this opens out as the cylin-

der bore enlarges. We prefer a cut in this shape:

Q. What principles should be carried out in regard to designing and making piston heads?

A. "The piston head should be deep enough to insure tightness, durability and good alignment, without being unduly heavy, especially in vertical engines. It is a good plan to make it of two portions cast from the same pattern, but one having turned on it a hub corresponding to a recess in the other. These two may be bored parallel, smaller than the rod, which is to be turned down parallel to fit the hole accurately, care being taken to leave no square corner at the junction of the full-sized and the reduced portions. It is cheaper to bore parallel than to turn the rod taper and ream a taper hole.

"The ring should be cast larger than the head; three water grooves turned in it, and a sufficient portion cut out diagonally to bring it down to the right diameter when

sprung in to the recess turned for it in the head."—*Power*.

LUBRICATION.

Q. How should lubricants be regarded?

A. Lubricants should be regarded as money savers, not as necessary evils. They should not be selected because of cheapness per gallon, but by reason of fitness for the special purpose.

Q. Will good lubrication save steam and fuel, or only ensure cooler running and greater durability of the engine?

A. I have for some years been trying to persuade the average lunkhead engine-owner and pig-headed "engineer" (courtesy title, in too many cases of which all know, for "starter and stopper") that there is no economy in first sloshing an engine all over, inside and out, with oil and then letting it run dry and cutting; that it takes more oil, more coal and more repair-bills, (to say nothing of time lost from shut downs) than where a decent lubricating device is used. But it is rarely that even with a good coal saving as a test I can prove the fact. 'Tis a fact all the same.

Q. What should be the characteristics of a good lubricant?

A. A lubricant should have body enough not to be squeezed out under great pressure; and yet be fluid enough to enter between tightly fitting bearing surfaces; should be a good conductor of heat, and have a high power of reducing friction. It should not be liable to decompose in the air or in use nor to corrode the bearings and journals, &c. It should vaporize at a high temperature and chill or thicken at a low; and of course be free from grit and adulteration.

Q. What are the best lubricants?

A. Sperm is the best general lubricant, but for a given friction-reducing power costs too much. It is generally better to pay half as much for a lubricant with two-thirds the anti-frictional and durable qualities that sperm has.

For light bearings the most fluid oil that will remain in place is the best.

Lubricants should adhere naturally to the bearing metal. In this respect mineral oils are first, then sperm, then neats-foot and then lard.

Q. What is the effect of heat on lubricants?

A. Bearings should never be allowed to *commence to heat,* as heating reduces the consistency of lubricants and causes them to squeeze out and cause still more heating, and so on.

Heating should be *prevented,* rather than cured. It cracks, distorts and abrades the surfaces in contact; and even sets fire to the lubricant; overheating, softening and weakening both the journal and the bearing, and even welding them together.

Q. Does it pay to use poor lubricants?

A. No. A saving of 10% in cost of lubricants has been shown by the indicator to cause an increase of 10% in the more important item of coal.

Q. What are the legitimate uses and proper modes of use of oil cans?

A. Oil cans have their legitimate uses and modes of use. They are not intended to squirt oil at an oil hole; but where it is necessary to oil through a small orifice the nose of the spout should be put in the hole, not simply near it. The "illuminating oil can" seemed to be one of the—the *very*

few—things which really "filled a long felt want;" but for some reason or other it has not got to general public knowledgo, much less into general public use.

Q. How may hot bearings be cooled down?

A. Sulphur and grease have a cooling tendency upon hot bearings. This is probably because the fine metallic dust formed by the hot journal combines with the sulphur to form a greasy sulphide.

Q. How should the guides and slides be oiled?

A. From cups on the cross head and also from stationary cups on the slides.

Q. What is a good way to oil crank pins and their brasses?

A. By having the crank pins bored and drilled for oil passage and connected with an oil tube parallel with the crank and extending as far as the center of the main shaft, where it has a filling cap. The faster the crank turns, the greater the centrifugal force tending to drive the oil from the closed end at the shaft center, to the crank pin end.

Another good way is by a wiper on the

crank, which licks a drop of oil from a metal or other strip, fed from an oil cup, every time the crank goes over.

Sometimes there are "oil cellars" attached to the under side of the straps; these being metal boxes filled with oil.

Q. How may the crank pin of a portable engine be kept oiled?

A. A simple arrangement for oiling the crank pin of portable engines is to make it of a steel tube, and fill it with tallow. By this means it may be made a perfect self oiler.

Q. How should oil be supplied to the cylinder?

A. Oil should be supplied to steam cylinders continuously and in small quantities. Attention to this will result in saving lubricant and in keeping the cylinder in better condition.

Q. What can be said of cylinder oils containing mineral oils?

A. Cylinder oils containing mineral oil are apt to be entirely free from all honeycombing action on the valve seat and cylinder wall; but when the piston has insufficient bearing surface, and the pressure is

great, they sometimes have not enough "body" to prevent cutting.

Q. What can be said of cylinder oils for high pressures and high speeds?

A. Cylinder oils for high pressures and speeds should be better than those for low. This is seldom the case. Tell your oil dealer what you need and see that he gives it to you.

Q. How should cylinder oils be tested?

A. The only way to test a cylinder oil is by an indicator, in the cylinder of an engine (preferably the one on which it is desired to use it) under the speeds and initial pressures of everyday use. Run the engine with some standard oil, as lard, tallow, &c., take friction cards and load cards; then run for a day with the oil to be tested, and then commence taking cards, with the same load on as before.

[The following query was made in reference to the question in the Catechism, concerning light piston head *versus* light cylinder head:]—

Q. Isn't there a misprint on page—of the Catechism where you state that a *piston*

head should be made light, so that, if it brought up against something, it would break instead of the cylinder head? This is contrary to locomotive and marine engineering. Some locomotive builders turn a narrow groove between the joint and bolt holes. Here on the Mississippi the flange on the head is made from $\frac{3}{8}''$ to $\frac{1}{2}''$ less in thickness than flange on cylinder. I think smashing the piston wrecks the engine more than smashing the head, because you can make a wooden cylinder head do, but a wooden piston would not last long.

A. The light head was advisedly recommended; for engines having late cut off because it has less inertia, hence reverses more easily than a heavy head; it should be cheaper to replace than a cylinder head; and the risk of the cylinder itself is something worth considering. It is a common thing in locomotive and stationary engines to have a heavy piston not only smash the cylinder-head but tear out part of the flange and wall of the cylinder proper. The narrow groove between joint and bolt holes is probably for the purpose of laying in a copper wire, to make a tight metal joint, or letting

a rubber gasket fill in. The flange on the head should be thinner than that on the cylinder, to cause the head flange to break rather than the cylinder; and the piston head should be lighter than either. A wooden piston would run for a while as well as a wooden head; the greater the diameter, the more this is so: and the consequences of running a wooden piston head are much less severe than those of blowing out a wooden cylinder head; particularly on shipboard, where there is more risk of scalding the engineers to death and causing a panic among passengers. A wooden head for the crank end of the cylinder would be difficult to get up and supply with stuffing box.

The following communications on this subject appeared in POWER for September 1885.

MESSRS EDITORS:—In reference to the subject, (page 153, of POWER, for August) of light piston heads *vs.* weak cylinder heads for quickness and cheapness of repair, in case of smashing by their coming together or getting water, nuts, etc., between them;—the question seems to be raised whether the cheapest of the two

parts named should not be made the weakest, and enough so to be certain to break in case of any of the accidents alluded to, and to break easily enough and completely enough to save the piston from all damage.

Unquestionably, it would be preferable that only one of the parts should be damaged rather than both, and also that the part broken should be the cheaper one. But unfortunately, the part most difficult to save by such means, namely the piston, is the most expensive one. If only water was to be provided for, it would be a matter of no difficulty to make the parts of such relative strength that the head would yield, but when a nut gets in it falls to the bottom and catches the piston where it is weakest and the head where it is the strongest, and if the latter were made so weak that the rod would not even be bent it would not be strong enough to properly resist the working pressure, still less those occasional whacks on a little water that impose considerably more strain on the parts than that due to the steam pressure, yet far from enough to do any damage to parts made reasonably strong. But since

water is by far the most frequent mischief maker, it may be asked, why not provide for it? To this it may be answered that unless the piston was made very massive and strong the degree of weakness in the head necessary to insure its breakage would be such that the legitimate stress from steam pressure with the help of the frequent light water concussions that all engines are liable to, would be sufficient to gradually "fatigue" the metal of the heads, that is, to gradually weaken it, so that it would, in time, give out from ordinary service. To avoid this "fatigue" is the function of the "factor of safety," well known to constructing engineers, which is generally placed at about five times the estimated breaking strength.

Still it may be said that given a piston with a factor of safety of, say ten, the head might have the required factor of safety of five and still be certain to break easily enough to save the piston, which is probably true. The difficulty in the way of obtaining the proper strength is that it would be almost impossible for the designer to determine it theoretically on paper, with

sufficient accuracy. Every different size and design of engine would have to have several of its heads broken by hydraulic pressure, enough, in fact, to ascertain not only the proper strength for a given quality of iron, but also the margin of uncertainty, arising from the unavoidable variation in the strength of the iron used.

A far better plan for such engines as contain distributing valves that will not yield to dangerous water pressures is to provide them with spring valves that will yield to a pressure slightly in excess of the maximum working pressure. Then neither piston nor cylinder head will be broken.

J. W. THOMPSON,
with Buckeye Engine Co.
Salem, Ohio.

Mr. Wm. Lee Church, M. E., formerly with the Buckeye Engine Co., and now of the firm of Westinghouse, Church, Kerr & Co., (Westinghouse and Reynolds-Corliss engines) says:—

"Personally I am of the opinion that less damage results in horizontal double acting engines from breaking the piston than from breaking the cylinder head. The piston is

apt to break up into small pieces and the subsequent movements of the rod under the action of the fly wheel will not usually do any damage. If the piston is made so strong as to wreck the cylinder head it is sometimes apt to crack the cylinder itself under the flanges. This will almost assuredly take place if the breakage occurs when the piston is moving towards the shaft. With the Buckeye engine we used to habitually make the piston the breaking piece; with a single acting engine, however, like the Westinghouse, all danger of breakage is necessarily on the up stroke, and we therefore make our pistons very strong and weaken the head artificially by a pop-out plug."

Mr. Wm. A. Harris takes the opposite view of the case; as may be seen by the following:—

MESSRS. EDITORS:—In my judgment, it is preferable to have a weak cylinder head in preference to the piston head, as in the latter case, if the piston was wrecked it would cut the cylinder, and very likely destroy it, while if the cylinder head was the weakest

and would give way, you would save your cylinder.

WILLIAM A. HARRIS.

Providence, R. I.

Q. Can any table be got up, showing the proportional average effective pressure in engine cylinders, when the point of cut-off is given?

A. No, because that varies with the clearance, the back pressure, etc., as well as with the point of cut-off, etc. Separate tables would have to be made for each clearance, for each initial pressure, and for each back pressure.

Cut off.	Act. Exp. Rate.	Hyp. Log.	Av. Tot. P. Above Vac.	Term. Pres. Above Vac.
.10	7.	1.946	0.2446	0.1428
.125	6.	1.792	0.2990	0.1667
.15	5.25	1.658	0.3487	0.1905
.20	4.2	1.435	0.4370	0.2381
.25	3.5	1.253	0.5133	0.2857
.30	3.	1.099	0.5797	0.3333
.333	2.741	1.007	0.6190	0.3393
.375	2.470	0.903	0.6640	0.4049
.40	2.333	0.846	0.6884	0.4286
.5	1.907	0.645	0.7725	0.5244

The table above gives the actual ex-

pansion rate, and its hyperbolic logarithm, the proportion of average total pressure above vacuum, and absolute terminal pressures, to initial pressure above vacuum, for each of several points of cut off, with 3% clearance at each end (engine non-condensing).

Q. Give full description of how you would proceed to set up a Cowdrey four cylinder engine if every piece was separated from the other.

A. Take the pistons, insert the pitmans in them, and pass the pins into the holes in the piston heads, then place the steam ring on the piston and screw the caps down firmly; place the pistons in the cylinders, holding the topmost one in place, by shoving the pin, that holds the pitman, out a little, so that the pin extends over the top of the cylinder, and then shove the shaft into the inner bearing; then bring the inner ends of the pitmans to their seat on the crank pin, and place the rings around them and bolt them. Then place the lower half of the drum cover in position, and loosely fasten it, then the upper portion, and likewise fasten it, then bolt the

two halves together, and then the whole cover to the drum. Then place the round valve upon the pin that projects into the steam chest, and place the cover on and securely fasten it. If the engine is reversible, place the reverse valve inside the rim or box, screw the lever on, place and bolt the cover, and after making the necessary connections to the boiler, set the oil cups to feeding properly, and the engine is ready to start.

Q. In what position, relative to the main crank, should the valve crank be to work steam full stroke?

A. At a right angle.

Q. How would you increase the lap?

A. By turning the valve pin away from the crank pin.

Q. How can the main, or round valve, be made to properly admit and release steam, if it has any lap?

A. By setting the valve pin so that the outside and inside of the valve expose an equal portion of the opposite ports.

Q. Why is it not necessary to fasten the pins that hold the pitmans in the piston heads?

A. Because the cylinders form guides and contain no openings which they must pass.

Q. One Cowdrey four cylinder engine is equal, in area, to how many engines of the ordinary type with same bore and stroke, and taking steam on both ends ?

A. Two.

Q. Describe the valve, and how it is balanced.

A. A disk with openings near its center, fitted with a steam packing ring, and enclosed on both its faces, and balanced by having steam on its circumference and there only.

Q. Describe the Cowdrey reversing valve, and how it is balanced.

A. A round ring fitted with a steam packing ring, and laid upon its side, and balanced by having steam only on the inside.

Q. State how the reverse valve produces a reverse motion of the Cowdrey engine.

A. By opening the former steam chest to the exhaust, and carrying steam to the inside of the valve, where it finds communication with the opposite cylinder,

whose piston is in mid stroke, checks its return to the end, or outside, of the cylinder, and sends it in the opposite direction.

Q. How would you increase the compression on a Cowdrey engine without altering the lead?

A. By increasing the thickness of the metal on the inside of the valve.

Q. How would you increase the lead of a Cowdrey engine without altering the compression?

A. By reducing the circumference of the valve.

Q. What is the Campbell exhaust condenser or exhaust head?

A. This device is intended for use with non-condensing engines as an exhaust-pipe head. It is intended to facilitate the condensation of the steam by providing a condensing chamber with an opening for the admission of air, so placed that the steam, when entering such chamber, will draw cold air in with it, which air, mingling with the steam, will cool it off and also cool the condensing surfaces, whereby the conden sation is greatly increased. Provision is made to collect and carry off the condensed

water without escape through the air inlet, thus depriving the steam of the moisture, and preventing the scattering of water or moisture on roofs and neighborhood, which causes so much damage and often becomes a nuisance. It is stated to relieve back pressure, and to produce great saving of feed-water by returning it. It is claimed that the escaping steam is dry and almost noiseless.

JACKETING.

Q. Suppose that there was just as much condensation in a steam jacket as it prevented in the cylinder, what would be the benefit of the jacket?

A. It would increase the power, or "capacity," of the engine.

Q. What should be done with the "condense" from the steam jacket?

A. It should be returned to the boiler at once, while hot.

Q. Do the pressure and temperature in the steam jacket vary during the stroke?

A. No.

Q. How is the exhaust jacket applied in compound engines in the British navy?

A. In the compound engine at present

so largely employed in the British navy, the cylinders, outwardly, are the same diameter, but the high-pressure cylinder, as previously mentioned, is generally made about ⅛ less in area than the condensing cylinder. The space round the actual high-pressure cylinder is used to receive its exhaust steam until the valve of condensing cylinder opens to admit it, as the pistons do not move simultaneously, owing to the cranks being at right angles to each other.

PRESSURES, TEMPERATURES, &C.

Q. In what way is steam "elastic?"

A. It is "elastic" when compressed, not when the volume of the vessel which contains it is increased. But it tends to act as a cushion spring if compressed.

Q. What is meant by steam being "condensible"?

A. It may be rapidly condensed (reduced to the form of water) by the sudden influence of a cooling body as water or a cold metallic surface, or it will slowly return to that state if left to radiate heat to the air through the walls of a

cylinder or pipe without receiving an additional supply of heat.

Q. What is dry steam?

A. Dry steam is that in which there is no unevaporated water, but which is no hotter than saturated steam, that is, no hotter than the boiling point corresponding to the pressure.

Q. What is the difference between a gas and a vapor?

A. A gas requires great pressure and very low temperature to reduce it to a liquid form; a vapor is liquefied at ordinary temperatures.

Q. What then is a vapor?

A. A vapor is any substance in the gaseous condition, at the greatest density consistent with that condition.

Q. Is there an "absolute zero" of temperature?

A. Reasoning fixes an "absolute zero" of temperature at that point where gaseous elasticity disappears=

—461.2° Fahr.

—274° Centigrade.

—219.2° Reaumur.

Q. What is specific heat?

A. The specific heat of a body is its capacity for heat; the quantity of heat required to raise its sensible temperature 1° F., as compared with the amount required to raise an equal weight of water from 32° F, to 33° F.

Q. What is a "heat unit"?

A. A British "heat unit" is the amount of heat required to raise one lb. of water from 32° F. to 33° F.

Q. Does it not take as much heat to raise one lb. of water from 32° F. to 33° F. as to raise it 1° at any other point of the scale?

A. No. It takes 1⅓% more heat, for instance, to raise one lb. of water from 212° F. to 213° F. than from 32° F. to 33° F.

Q. Does the specific heat of other substances than water vary with the temperature?

A. The specific heat of all so called "permanent gases" is practically constant for all temperatures and densities; the capacity for heat of any one gas, is about the same at all temperatures.

Q. Of what use are the "specific heats" of substances to us?

A. By knowing the specific heats of their constituents we may calculate the heating power of fuels, &c.

Q. Of what practical use is it to calculate the heating power of a fuel?

A. By getting an average sample of any fuel analyzed, the amount of steam at 200° F. which a pound thereof should make from water at 212° under theoretically perfect conditions can be calculated; and the amount of air needed, can be ascertained. Then the valve of one fuel found, compared with that of another; and that necessary to perfection of action of the boiler be found out.

Q. What is latent heat?

A. Latent heat is heat which has disappeared or lies hidden. For instance, in evaporating one lb. of water at 212° F. and under a pressure of one atmosphere=14.7 lbs., 966.1 heat units disappear become latent.*

Q. Of what practical use is a knowledge of the latent heat of water?

A. Among other applications I may mention reducing the evaporation-rate of a boiler to a standard from and at 212°.

*Regnault.

Thus having obtained the total heat of evaporation at the actual temperatures of feed water and of steam, dividing that total heat by 966 gives a multiplier by which the weight of water actually evaporated by each pound of fuel is to be multiplied to reduce it to the equivalent evaporation from and at 212°; that is, the weight of water which would have been evaporated by each pound of fuel had the water been both supplied and evaporated at the boiling point, corresponding to the mean atmospheric pressure.

Q. How much heat does it take to generate one pound of steam?

A. That depends on the temperature of the water and the pressure of the steam. The colder the water and the higher the pressure, the more heat required. To make a pound of steam at 212° F. = 100° C., from one pound of water at 32° F. = 0° C., takes three separate lots of heat, as follows: (1) To raise the water from 32° F. = 0° C. to 212° F. = 100 C., it takes 180.9 heat units. (2) To make that water into steam *at no pressure,* 892.9 heat units; and (3) To raise that steam at no pressure to steam

at the atmospheric pressure of about 14.7 lbs. per square inch above vacuum, 72.3 heat units. Total, 1146.1 heat units, equal to 883,789 foot pounds of work.

Q. Are all bodies expanded by heat?

A. Yes, but in different proportions.

Q. How many kinds of expansion are there?

A. Three directions in length, in breadth and in thickness; also three kinds; linear in one direction; superficial, in two directions, and cubical in three directions. Superficial expansion is almost exactly twice the linear expansion; cubical expansion is almost exactly three times the linear expansion.

Q. How much does a hollow body expand?

A. A hollow body expands just as though it were a solid, having the same outside dimensions.

Q. Is the expansion of solids uniform?

A. The expansion of solids between 32° F. and 212° F. is sensibly uniform.

Q. What is the mo t expansible metal?

A. Zinc.

Q. How much does wrought iron expand?

A. Wrought iron expands about $\frac{1}{7}$ or $\frac{1}{8}$ per cent; (that is, from 1 in 700 to 1 in 800) when heated from 32° to 212° F.

Q. How does heat affect the volume of water?

A. Water on being heated from 30° to 39.1 F. contracts; then expands until at 46° F. it has the same volume as at 32° F.; then expands until 212°, the boiling point under one atmosphere; thence continues to expand as it is heated.

Q. How much does water expand between 32° F. and 212° F.?

A. Between 32° F. and 212° F. water expands .0465; that is, between $\frac{1}{21}$ and $\frac{1}{22}$ its volume at 32° F.

Q. Does the expansion of water take place at the same rate as the increase of temperature?

A. The volume increases in a greater ratio than the temperature.

Q. How can one find the density of water at a given temperature?

A. To find the density of water at a given temperature, (approximately): To the given temperature in degrees F., add 461, and divide the sum by 500. Again, divide

500 by that sum. Add the two quotients and divide 124.85 by their sum.

Thus:—The density of water at 200° F.

$$\frac{200+461}{500} = \frac{661}{500} = 1.325;$$

$$\frac{500}{200+461} = \frac{500}{661} = 0.7564;$$

$$1.325+0.7564=2.0814;$$

$$124.85 \div 2.0814=59.98=$$

weight of one cubic foot, at 200°, in pounds.

Q. How can the volume of feed water to produce a given volume of steam, be ascertained?

A. By adding 458 to the steam temperature, multiplying the sum by 373 and dividing the product by the steam pressure in pounds per square inch.

Q. How can we find the admission period requisite for an actual expansion rate?

A. To get the required admission period for a given actual expansion rate—divide the length of stroke, plus clearance, by the actual expansion rate, and deduct the clearance from the quotient.

Thus an engine of 24″ stroke, 5%(=

0.1 foot) clearance, would, to get an actual expansion rate of 4, have to cut off at

$$\frac{2.10}{4} - 1 = .4225.$$

Q. What principle should be carried out in regard to designing and making piston heads?

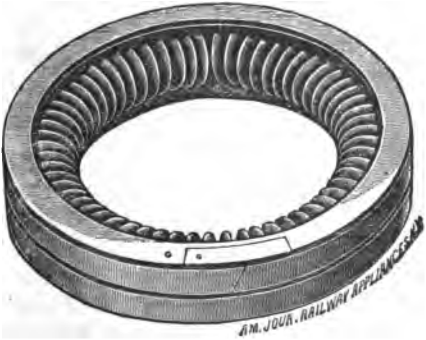
A. "The piston head should be deep enough to insure tightness, durability and good alignment, without being unduly heavy, especially in vertical engines. It is a good plan to make it of two portions cast from the same pattern, but one having turned on it a hub corresponding to a recess in the other. These two may be bored parallel, smaller than the rod, which is to be turned down parallel to fit the hole accurately, care being taken to leave no square corner at the junction of the full-sized and the reduced portions. It is cheaper to bore parallel than to turn the rod taper and ream a taper hole.

"The ring should be cast larger than the head, three water grooves turned in it, and a sufficient portion cut out diagonally to bring it down to the right diameter when

sprung in to the recess turned for it in the head."—*Power*.

Q. What is the Lancaster & Tonge piston?

A. In this piston, a straight spiral spring, bent into a circle, forces the rings out against the walls of the cylinder, and at the



LANCASTER & TONGE PISTON PACKING.

same time holds them firmly between the junk ring and the piston body. The pressure between the piston rings and the cylinder results from the natural elasticity of the spring and its effort to regain its natural shape, and does not depend upon the screw-

ing down of the junk ring. There is claimed to be no danger of excessive friction being set up between the piston and the cylinder by careless adjustment.

Q. Where should the cross-head pin be?

A. In the center of length of the slides, so as to give straight thrust of these on the guides.

Q. When does an engine "run over?"

A. An engine "runs over" when, if you stand at the cylinder and look towards the fly wheel, the top of the fly wheel runs from you; and *vice versâ*.

Q. How should the crank be fastened to the shaft?

A. It should be both pressed and keyed on.

Q. What is the duty of the fly wheel?

A. The duty of the fly wheel is to store up work during one stroke or portion of a stroke, and give it out to the crank during another period, when the demand is greater.

Q. What class of engines demands the heaviest fly wheels?

A. The more irregular the work, the

heavier the fly-wheel must be, to ensure safety to the engine and continuance of the work. The more delicate the work, and the more exacting of regularity of speed, the greater the desirability of a heavy fly wheel, to ensure even quality of the work.

Rolling mills, in which the work is very intermittent and irregular, come under the first head; and paper and flouring mills, cotton spinning factories, and "dynamo" machines for electric lighting, come in the second class.

Q. What rules are there as to the quantity of work to be stored up in the fly-wheel?

A. Watt called for a storage of the work during 7½ strokes, Bourne says 6.

Q. What may be said about centering and balancing the fly-wheel?

A. The fly-wheel must be accurately centered and balanced, especially where it is a pulley-wheel driving smaller pulleys, at higher speeds, in which case its irregularity would be multiplied.

Q. What limit may be assigned for fly-wheel speed?

A. One rule puts 80 feet of rim speed

per second, equals 4800 feet per minute, as the maximum rim speed for a fly-wheel. This would correspond to about 95 turns per minute for a 16 foot wheel.

Q. What is the usual mean radius of the fly-wheel ?

A. The usual mean radius of the fly-wheel is from three to five times the crank length (Rankine).

Q. What may be said about fly-wheel pits ?

A. These should be provided with good drainage, as wet main-belts are not desirable.

Q. Should fly-wheels be left unguarded ?

A. No. They should be caged in to prevent accidents by men or materials being thrown against them, or getting in the pit.

Q. Which need the the heaviest fly-wheels; automatic or throttling engines, and why ?

A. Automatic engines need the heaviest fly-wheels, for a given rate of speed and average pressure, because the pressure in the cylinder during one stroke varies more than in throttling engines.

Q. Which needs the heaviest fly-wheels,

an engine which cuts off early, or one which cuts off late; and why?

A. The early cut off takes the heaviest fly-wheel for a given average pressure and given rate of speed, because the cylinder pressures vary more during a stroke.

Q. If two engines have the same piston speed and one has double the number of revolutions the other one has, will they have the same tendency to shake?

A. No. The accelerating force will be twice as much in the former case; and the piston come to its maximum speed in half the time, and in doing it will move over only half the space.

Q. How can an inverted vertical engine be balanced?

A. By using a supplementary steam balance cylinder, above the main cylinder, to annihilate the dead weight.*

Q. How may a marine engine be got off its center quickly and without danger?

A. By having a V-grooved friction wheel on the shaft, and opposite to this placing a short straight block of iron grooved to

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* See paper of W. S. Durfee, Cleveland meeting Am. Soc. M. E., 1883.

correspond. This block is connected to the piston of a short stroke steam cylinder and arranged so as to be made to move towards the friction wheel, and touch it, by an eccentric journal.*

Q. What is the leverage of friction of a cylindrical journal?

A. The leverage of friction of a cylindrical journal is the radius of the journal.

Q. How can horizontal stationary engines best be mounted for testing balance?

A. B. W. Payne & Sons, in testing the balance of a horizontal stationary engine, put the engine on rollers, cross-wise with the cylinder axis, and connected with it by a flexible steam pipe. There are few engines that will stand this sort of test, and few men who can bring any single cylinder engine up to the standard imposed thereby, of running without traveling to and fro on the rollers.

Q. How may the weight of a well proportioned engine be approximated?

A. By the cubic foot of cylinder. The following figures are from Haswell:—

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* J. F. Holloway, in a paper before the A. S. M. E., July, 1883.

WEIGHTS OF STEAM ENGINES. Side-wheels.—American Marine (Condensing).

Engine.	Frame.	Water-wheels.	No. of Cylinders.	Volume of Cylinders.	Weight per cubic foot.	Service.
Vertical beam...	Wood.*	Wood.	1	63. cu. ft.	1100 lbs.	River.
do. ...	Wood.*	Wood.	2	216. "	1040† "	Coast.
do. ...	Wood.*	Iron.	1	530. "	1500 "	Sea.
do. ...	Wood.*	Iron.	1	725. "	1089† "	Sea.
Steeple..........	Iron.	Iron.	1	12.8 "	3800 "	River and Coast.
Oscillating......	Iron.	Iron.	2	540. "	850 "	Sea.
do.	Iron.	Iron.	2	1090. "	760‡ "	Sea.
Inclined direct...	Iron.	Iron.	2	534.5 "	1100 "	U. S. Navy.

* Without frame. † With frame 1109. ‡ Single frame.

English, per Nominal H. P.

Oscillating condensing engines	560 lbs
Wheels	246 "
Boilers	481 "
Coal bunkers	67 "
Water in boilers	224 "

Screw Propellers —Marine (Condensing).

Engine.	No. of Cyl.	Vol. of Cyl.	Weight per cubic foot.	Service.
Vertical direct	2	12.5 cu. ft.	5600 lbs.	Sea.
do.	1	69. "	4660 "	Sea.
do	1	88. "	8650 "	Coast.
Vertical compound, single	2	215. "	2200 "	Sea.
do. double	4	506. "	8010 "	Sea.
Trunk	2	193. "	2080 "	Sea.
Horizontal back-action	2	68. "	4260 "	Sea.

Non-condensing Engine.	No. of Cyl.	Vol. of Cyl.	Weight per cubic foot.	Service.
Horizontal direct	2	3.9 cu. ft.	4990 lbs.	River.
Vertical direct	1	9.8 "	7800 "	Coast.
Inclined direct	2	4. "	5200 "	Coast.

Non-condensing Land-engines.	Engine.	Spur-wheel and connections.	Sugar Mill complete.	Boilers, Grates, etc.	Engine per Cube Foot of cylinder.
Vertical beam, 18 in. × 4 ft.	67,200 lbs.	87,800 lbs.	89,600 lbs.	26,880 lbs.	9600 lbs.
do. 80 in. × 5 ft.	105,000 "	137,159 "	265,879 "	75,000 "	9884 "
Horizontal, 14 in. × 2 ft.	10,914 "	—	—	8,200 "	5100 "
do. 22 in. × 5 ft.	56,000 "	—	—	80,140 "	5600 "

Q. How should bed-plates be leveled up?

A. Bed-plates should be leveled up with sulphur, Portland cement, or lead, their values for this purpose being in the order named.

Q. How can grouting be got in large thin spaces between bed-plates and foundations?

A. By using a stand-pipe through which the cement is introduced.

Q. How may holes be tapped square in cylinder flanges?

A. To tap holes square in cylinder flanges, etc., it is best to have a gauge piece composed of a flat piece of wrought iron having at one end a hub which is bored true square with the flat end, and which has steel bushes or thimbles to fit different taps. A centering plug is employed where desirable, to set the gauge piece over the center of the hole.

WEIGHTS, DENSITIES, FACTORS, &c.

Q. How much does a gallon of pure water weigh?

A. One imperial gallon of pure water at 62° F. has a volume of 277.123 cubic inches and the weight of 10 lbs. But the British

act of 1825, often considered as in force yet, gives it 277.274 cubic inches, and 10.00545 lbs.

Q. How much does one cubic foot of pure water weigh?

A. One cubic foot of water weighs at

32° F = 0°C.	62.418 lbs.
39.1	62.425.
62° (Standard temp.)	62.355.
212° F=100° C.	59.640.

(*D. K. Clark.*)

Q. What is the pressure of a column of water 1 ft. high?

A. A column of water at 62° F. and 1 ft. high exerts a pressure of 0.433 lbs. per square inch on its base.

Q. What height of water column exerts a pressure of 1 lb. per square inch on its base?

A. A column of pure water at 52° F. and 2.3093 ft.=27.71 inches high, exerts a pressure of one pound per square inch on its base. Thus 50 lbs. per square inch would be exerted by a column of pure water $2.3093 \times 50 = 115.465$ ft. high.

Q. How much does 1 cubic foot of sea water weigh?

A. One cubic foot of average sea water

at 62° F. weighs 64 lbs. or 40-39ths, or 1026-1000ths as much as fresh water. 35 cubic ft. of sea water weigh 1 gross ton.

Q. How much solid matter in sea water?

A. Sea water contains 3.40 lbs. of solid matter in 100 lbs., that is, $\frac{1}{30}$th part by weight.

Q. What is the weight of 1 cubic foot of air?

A. One cubic foot of air, under a pressure of 1 atmosphere or 14.7 lbs. per square inch, weighs at 32° F., .080728 lbs=1.29 oz.= 565.1 gr. At 62° F. it weighs .076097 lbs. =1.217 oz.=532.7 gr.

Q. One lb. per sq. inch is how many lbs. per sq. ft.?

A. One lb. per sq. inch is 144 lbs. per sq. foot. It is also 2.0355 inches of mercury at 32° F. or 2.308 ft. of water at 52.3° F.

Q. One lb. of fuel per hour per H. P.= how many foot pounds per lb. of fuel?

A. One lb. of fuel per hour per H. P. is 1,980,000 foot pounds per lb. of fuel, or 2565 units of heat.

Q. 1,000,000 ft. lbs. per lb. of fuel is how many lbs. of fuel per hour per H. P.?

A. 1.98 lbs.

Q. Give the weights and specific gravities of the various materials used about a steam engine.

Materials.	Wt. of 1 cu. ft.	Specific Grav.
Lead sheet	712	11.418
Copper "	547	8.805
Bronze:		
84 cop. 16 tin	534	8.56
81 " 19 "	528	8.46
Brass, cast:		
75 cop. 25 zinc, sheet	527	8.45
66 " 34 "	518	8.30
Steel (least and greatest density)	435 493	7.729 7.904
Iron, wrought, least and greatest density	466 487	7.47 7 808
Iron, cast, least and greatest density	378.25 467.66	6 9 7.5
Tin	462	7.409
Zinc, sheet	419	7.2
" cast	428	6.86
Limestone	178.9	2 58
Granite, gray	180	2 9
Sandstone	130-150	2.08-2.5
Concrete:		
Portland cement 1 to gravel 10	139	2.23
P. cement 1, sand 2	127	2.04
Mortar	109	1.75
Bricks	124.7-135.3	2-2.17
Brickwork	110-115	1.76-1.84
Masonry, rubble	115.3-143 4	1.85-2.3
Sulphur	124.7	2
Coal, Anthracite	85.4-99.1	1.37-1.59
" Bituminous	74.8-81.7	1.2 1.31
" Cannel	74 8	1.2
Lignum vitæ	40.5-82.9	.65-1.33
Oak, American red	54.2	.87
Pine, American	21	0.34
Charcoal (av.) in powder	93.5	1.5
" " in sm. pieces, heaped	25.3	0.405
India rubber, pure	58.	0.93

THE STEPHENSON LINK.*

Q. With the Stephenson link motion, how may the lead and cut-off be equalled?

A. They cannot, with a circular curve in the link, be made the same for both ends of the cylinder, for all points of cut-off, and for both front and back gear.

Q. What then is done in this connection?

A. The back gear is disregarded.

Q. Can the lead and the cut-off be equalized for the forward gear alone?

A. No; not for all points of cut-off, with the ordinary circular curve for the link.

Q. What then is done in this connection?

A. The points of cut-off are made the same for ½ gear forwards, because that is the position in which the link is most often used.

Q. How may we change the times of admission for the front and back ends of the cylinder most, in relation to each other?

A. By changing the position of the point of suspension on the links.

---

* See also pages 34, 40, of this volume; and page 145 of Part I. For other items on link motion, see pages 34, 35, 37, 41, 44, of this volume.

Q. How may this be done?

A. By moving its point up and down, or vertically.

Q. Where are the links generally suspended?

A. From a point about half way between the eccentric rod connections, and about $\frac{1}{4}$ to $\frac{3}{4}$ inch back of the centre line of the link slot.

Q. Would there be any better point of suspension, as regards steam distribution?

A. Yes, about three inches above the centre.

Q. Why not hang it there, then?

A. Because this would, upon a locomotive at least, necessitate a shorter link hanger; and this would be liable to be too much strained by the motion of the link.

Q. How can you set a Stephenson cut-off so as to cut off at seven feet stroke?

A. I cannot answer your question as to the link motion unless I know the arrangement of parts. If I had that and the dimensions, or rather distance between all centres, I could calculate it for you and give you the answer. Failing that—if I had a rough sketch showing arrangement

of parts, I could give you a rule so you could fill in dimensions and calculate, or perhaps even so you could put the lever right at the point without calculation.

Q. What is the forward or "out" stroke?

A. That made towards the crank.

Q. How can you find the point of exact half-throw of the crank?

A. Square up from the crank centre, at right angles to the guides, or, if the engine be level, "plumb" up from the crank centre.

Q. How can you find the length of the connecting rod?

A. By putting the piston at mid-stroke and measuring from the centre of the cross-head pin to the centre of the shaft.

Q. How do you tell the distance from the centre of the cross-head to the centre of the crank shaft?

A. When the piston is at half stroke, then the distance between these two points equals the length of the connecting rod. At any other point take from the length of the connecting rod the distance of the piston from the inside of the crank end of the cylinder, and add to this half the stroke.

Q. How would you find the length of the eccentric rod of a link motion?

A. Put the valve upon its half stroke, and measure from the centre of the eye to the centre of the shaft; then take off half the shaft diameter, and the thickness of the strap under the foot of the rod.

Q. In an oscillating engine, how can you tell whether the centre of the trunnions and the centre of the crank shaft be fair?

A. Turn the engine over and see that the crank pin brass is always the same distance from the cheek of the crank.

THE ECCENTRIC.*

Q. How may the throw of an eccentric be measured?

A. Subtract the height of the low side from that of the high. Thus:—18″ eccentric, 3″ out of the centre, on a 6″ shaft, has a 3″ low side, and a 9″ high side; 9″ − 3″ = 6″ = total throw, or double the eccentricity.

Q. What will show when the eccentric has slipped?

A. If the engine runs, it will be noted

* See also pages 26, 28, 34, 38, 39, 40, 41, 109, 110, 122, 128, of this volume; and pages 36, 46, 55, 106, 146, 158, of Part I.

that the exhausts from the crank and out ends are not the same.

Q. If the forward-motion eccentric broke, could the engine be worked ?

A. Yes; by slinging the backing eccentric in place; but in that case the engine could not be run in the backing motion.

Q. What is a loose eccentric ?

A. One having balance weights and stops, used for reversing engines not having regular reversing gear.

Q. How is an engine reversed that has a loose eccentric ?

A. The eccentric rods are thrown out of gear, and the slide valve put by hand in mid-travel ; then the valve is worked by hand in the direction to reverse the engine; the eccentrics do not turn until the stops get against them; then the eccentric rods are thrown in again.

MEASURING, SQUARING, ETC.

Q. What should be done, when a marine engine is set up, to facilitate setting the cut-off for any given point ?

A. It should have prick-punch marks made on the valve stem or some other moving part of the valve gear, showing when

the valve is "line and line" at each end, just about to admit steam; also similar marks as to cut-off, exhaust release, and cushion, where there *is* cushion.

Then the course of the cross-head can be very readily laid down on the guides, and, the piston being held at seven feet stroke, or wherever you wanted to cut off, the valve gear could be put at that notch or clamped at that place, which will bring the "cut-off" witness-mark right for cutting off. The quadrant should then be marked, so that you can get the place the next time without delay or trouble.

Q. What about equalizing the cut-off in marine or other engines having link motion?

A. With most gears, if you cut off at seven feet, or any point on one end, you will cut off earlier or later on the other, unless you sacrifice equality of lead. Generally if you get the cut-offs "squared" for one point of cut-off, the engine will cut off unevenly with any other grade of expansion.

Q. When should the work of marking and setting be done?

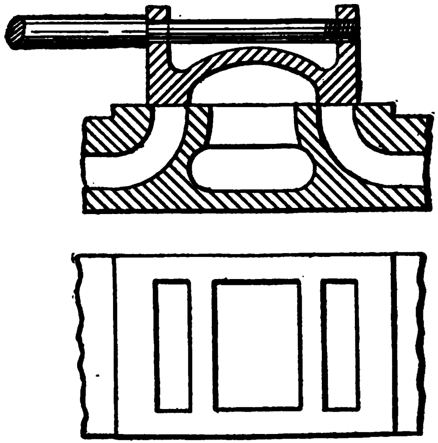
A. While the engine is hot all through;

as an engine set cold will not perform the same when hot.

THE SLIDE VALVE.*

Q. What are the recesses in a valve seat for?

A. Partly to prevent the valve wearing convex so soon, but more especially to pre-



LAPLESS VALVE AND RECESSED SEAT.

* See also pages 23, 31, 32, 108, 110, 111, 118, 122, 131, 166, of this volume; and pages 31, 32, 33, 71, 164, 174, of Part I.

vent a shoulder being worn by the steam edge of the valve.

Q. How far from the steam edges of the ports should they be ?

A. Close enough to the outer edges of the end ports to just let the steam edges of the valves pass them; then there can be no danger of shoulders wearing on the seat.

Q. Can a slide valve be absolutely lapless ?

A. No; there must of course be enough lap to make a steam-tight "seal" at each end when the valve is at mid-position, otherwise the engine would soon "blow through."

Q. Which is the worse, too tight a piston packing or too loose ? and why ?

A. Too tight; because there is danger of cutting the cylinders.

Q. What changes take place in the lead of an inverted vertical engine, as it wears ?

A. The top lead increases, and the bottom lead decreases.

Q. Where a slide valve has variable travel, how is it likely to wear ?

A. The valve is liable to wear convex.

Q. If the amount of lap that an ordinary

slide valve has, be altered, should there be any other change? If so, in what?

A. Yes; the position of the eccentric should be changed.

Q. What should this change be in case some of the lap was cut off?

A. The eccentric should have less angular advance; that is, it should be turned upon the shaft in the opposite direction to its regular forward motion, until the steam edge of the valve has just the desired amount of lineal lead when the crank is upon the centre.

Q. What would be one great objection to a slide valve without lap, outside of the question of economy?

A. Danger of the engine sticking upon the centre, if it were a marine engine.

Q. "How much lap is required for a slide valve having $3\frac{1}{8}''$ travel, to cut off at $\frac{5}{8}$ stroke of piston?"

A. If there is no lead, $\frac{15}{16}$ inch.

Q. Why is it that most types of automatic cut-off engines cannot cut off later than $\frac{5}{8}$ or $\frac{3}{4}$ stroke as a maximum?

A. Where there is a main valve and a riding cut-off valve, the main valve usually has

steam lap proportioned to cut off at some definite point, if the cut-off valve does not effect cut-off sooner; and when the cut-off is by a "release gear," the valve has lap which will cause cut-off at some stated point if the trip does not operate before that time. With most release gears the governor can only cause cut-off while the valve is opening; and this is generally during less than half the stroke.

Q. "Do you recommend inside lap on an engine running at one hundred revolutions or more per minute?"

A. "100 or more turns" means 100 or 1,000, or almost anything, and the piston speed has an influence also. The amount of inside lap depends on too many things to be answered in general. Where there is considerable lead there may be no need of inside lap at even quite high speeds.

The Steam Engine Catechism, Part I., gives considerable information as regards the influence of inside lap, pages 37, 49, 51, 52, 53, 54; concerning lead, pages 41, 43, 51, 55, 125; port area, page 114; clearance, 28, 66, 81, 83.

Q. Can the D form of valve having no

outside lap, employed in automatic engines, work successfully ?

A. The D valve without outside lap cannot cut off nor work successfully unless helped by a riding cut-off valve, or by some other independent cut-off attachment, such as the "Tremper" rig. Of course with a good throttling governor it would work as automatically as any throttling engine; but not economically.

In fact, if the engine was overloaded, a lapless D valve with a good throttling governor would work more automatically than if it had lap, and had to cut off whether or not it was necessary. But it would not, of course, work economically without expansion.

Q. In what case is an Allen valve the most efficient ?

A. Where the valve travel is very short, and the point of cut-off very early.

Q. "Is not more power used up in *moving* double valves, than is saved in steam by the short feed ports ?"

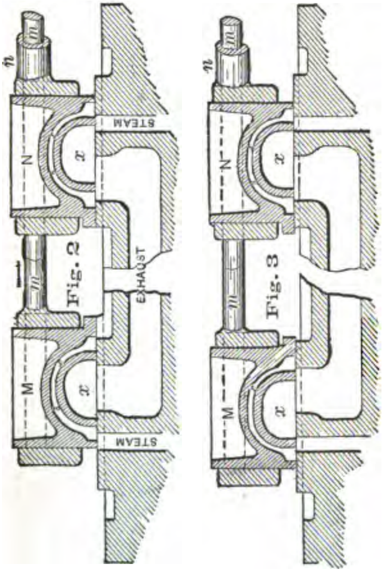
A. The amount of power required to move slide valves, either single or double, is almost an absolutely unknown quantity, and it would be doubly difficult to compare the

amount of steam needed to overcome it, with such a variable quantity as the amount wasted in the clearance spaces, as this last depends on the amount of cushion on other things.

Q. "Why do many engine builders who make double-valve engines, make the exhaust ports so narrow? I contend that the exhaust ports on *all* engines using the D valve, whether single or double, should be twice the width of feed ports, and if a trifle over would do no harm."

A. Most exhaust ports are too narrow, on all classes of engines; double the area of the admission ports would be best for any engine; of course counting the amount of area *uncovered.* If, however, the exhaust ports are large enough to run the engine without back pressure, at highest pressure, latest cut-off and greatest speed, without exhaust release, there is no harm in having the steam ports still larger, particularly where, as with double ported engines (*i.e.*, those having one admission port at each end), the passages are so short that a large cross section of port and passage does not give excessive cushion. Engines with early cut-off need

all the chance that they can get to get the steam in; particularly if they have what is erroneously called "negative lead."



STEVENS VALVES.

Q. What are the Stevens valves?

A. One variety thereof is very clearly shown in two positions in the annexed cuts,

and the action of the steam edges, both internal and external, may very readily be analyzed by marking a card to correspond with the valve, and moving it to and fro over the seat. In another style the valves have two passages through them; one for admitting and one for exhausting steam to and from the cylinder ; the steam is also admitted and exhausted in the usual way. (These valves are upon the Wilson principle.) Another is shown in the cuts here given. This resembles somewhat the Allen valve, with this difference—steam is admitted and exhausted to and from the cylinder through the same passage. This valve also takes steam and exhausts in the usual way, so that the steam is got into the cylinder doubly and exhausted doubly.

Q. A cylinder port is 18 inches long, 2½ inches wide; the steam lap of the valve is 1 inch, and its travel 6 inches. What is the greatest area that the valve is open for steam?

A. One-half travel, 3 inches; from this take the lap, 1 inch; we have left 2 inches, which is the amount open to steam; this, multiplied by the port length, 18 inches, gives 36 square inches.

Q. How do you find the greatest area that a slide valve is open for steam ?

A. Take the steam lap from $\frac{1}{2}$ the actual travel; multiply that by the length of the port.

Q. How can you tell the amount that the exhaust port of a common slide valve is open when the piston is at stroke end ?

A. Add the steam lap to the lead; take from this the exhaust lap.

Q. What is the general rule for uncovered port area ?

A. Port area uncovered should at various portions of the stroke be proportioned to piston velocity at that portion of the stroke.

Q. What should be the steam port area ?

A. D. K. Clark gives as the requisite steam port area, for 600 feet piston speed per minute, $\frac{1}{10}$ the piston area; and for other piston speeds in proportion. This gives us the table on next page of steam port areas for various piston speeds and diameters.

This table can easily be expressed graphically.

Q. How may the various amounts of port opening with the ordinary slide valve in different positions be most easily studied ?

Cyl. Diam. inches.	Cyl. Area Sq. inches.	Piston Speed, Feet per Minute.				
		500 .0833	600 .10	700 .11667	800 .13333	1000 .16667
10	78.54	6.55	7.85	9.16	10.47	13.1
12	113.097	9.43	11.31	13.20	15.07	18.85
14	153.938	12.83	15.39	17.96	20.52	25.75
16	201.062	16.76	20.11	23.46	26.81	33.51
18	226 980	18.92	22.70	26.49	30 26	37.83
20	314.16	26.18	31.42	36.65	41.89	52.36
24	452.39	37.69	45.24	52.78	60.32	75.40
30	706.86	58.91	70.69	82.49	94.25	117.81
36	1017.87	84.82	101.79	118.79	135.71	169.65
42	1385.44	115.45	138.54	161.68	184.72	230.91
48	1809.56	150.79	180.96	211.12	241.27	301.60
60	2827.44	235.62	282.74	332.96	376.98	480.12
72	4071.51	339.29	407.15	475.02	542.85	678.60

A. By marking the seat and ports to scale on the top edge of one straight strip of pasteboard or wood, and the valve face on the bottom edge of another strip, marking the centre of each, and on the seat the points of extreme throw of the valve, and then noting how much each port is opened or closed at each position of the valve.

Q. If the length (or width) of the lips of a slide valve is not given, how is it ciphered out from the other dimensions?

A. Add the two laps to the steam port

width. Thus: If a valve has $\frac{7}{8}$ inch outside lap and $\frac{1}{8}$ inch inside lap, and the steam port is $1\frac{1}{4}$ inches, the lips (or "legs," as some call them) are $1\frac{1}{4} + \frac{7}{8} + \frac{1}{8} = 2\frac{1}{4}$ inches each.

Q. How can you get the length of the arch of a slide valve when you cannot get at it to measure it?

A. Add to the width of exhaust port the width of the two bridges, and subtract the width of the two inside laps. Thus: Suppose exhaust port $2\frac{1}{2}$ inches; bridges, each, $1\frac{1}{8}$ inches; inside lap $\frac{1}{8}$ inch at each end; then the arch is $2\frac{1}{2} + 1\frac{1}{8} + 1\frac{1}{8} - \frac{1}{4} = 4\frac{1}{2}$ inches.

Q. Please state the proper proportions used by standard builders for the slide valves of a 19 × 24 passenger locomotive for high speed.

A. I have not at hand many figures for a 19 × 24; but suppose that these, taken from Cooke engines on the D., L. & W. R. R. coal-burning engines, would answer:

Steam ports	$1\frac{1}{4}''$ each
Bridges	$1\frac{1}{8}''$ "
Exhaust port	$2\frac{1}{2}''$
Valve arch	$4\frac{3}{4}''$
Steam lap	$\frac{3}{4}''$ each end.
Exhaust lap	none.

I would recommend $\frac{1}{32}$ inch exhaust lap

each side, which would bring the valve arch down to $4\frac{1}{16}$ inches. I would make the ports at least 18″ long.

Q. What should be the travel of a lapless valve?

A. A lapless valve should have for travel double the steam port width.

Q. What should be the travel of a lapped valve?

A. A lapped valve should have for travel at least twice the sum of the widths of leg and of steam port.

Q. What effect on the cut-off has increasing the travel?

A. Increasing the travel very slightly retards the cut-off.

Q. What is "over-travelling"?

A. Over-travelling is where the steam edge of the valve passes the end port and travels over on to the bar or bridge.

Q. What is the effect of having so little travel as to open the steam port only part way?

A. If the travel is so short as to only partially uncover the steam port, the steam is apt to be wire-drawn, at least at high piston speeds.

Q. Does a plain D slide valve move with the piston of a steam engine, or against its motion ?

A. It does both.

Q. When the piston is at the back end of the cylinder, where is the valve, supposing that there is no rocker arm and that the engine has not negative lead ?

A. It has moved quite a little distance forward, so that the back steam edge of the valve is just uncovering the back edge of the back steam port.

Q. When the piston has got well toward the centre of its stroke, what is the valve's action ?

A. It commences to move in the opposite direction to the piston.

Q. When the piston is very nearly at front stroke end, where is the valve ?

A. It has commenced to open the front port for the purpose of admitting steam to the front of the piston.

Q. Up to the end of the stroke, how does the valve run ?

A. In the opposite direction to the piston.

Q. When the piston reverses to go on the back stroke, what is the valve movement ?

A. They travel in the same direction (backwards) until the valve has made its full back stroke.

Q. What then?

A. The valve reverses and goes forward, running forward until the piston is at back stroke (engine on back centre).

Q. What next?

A. The piston reverses and runs the same way as the valve, as on the preceding forward stroke.

Q. Lay this down for a 12″ stroke engine.

A.

PISTON RUNNING FORWARD.		VALVE RUNNING.
Distance travelled	0″	Forward.
	3	(Still.)
	6	Backward.
	9	"
	12	"
PISTON RUNNING BACKWARD.		
Distance travelled	0″	Backward.
	3	(Still.)
	6	Forward.
	9	"
	12	"

Q. How may you note the actual and relative motions of piston and valve?

A. By watching the rods.

Q. Why should every engineer do this ?

A. Because some motions open the valve suddenly and close it suddenly, and some open and close too leisurely.

Q. Does the steam chest always contain live steam ?

A. The steam chest in the "Buckeye" engine contains exhaust steam only.

Q. Is an absolutely balanced valve desirable ?

A. There are many who claim that an absolutely balanced valve would not be desirable, because enough friction to keep the surfaces bright and prevent corrosion from starting leakage is necessary.

Q. What is the rule ordinarily given for the pressure which tends to force a slide valve to its seat ?

A. Add the area of one steam port to the area of the valve port, and multiply these by the mean effective pressure per square inch.

Q. Does the pressure in the steam chest (tending to press an unbalanced slide valve to its seat) vary during the stroke ? If so, why ?

A. "Steam chest diagrams," taken by the

indicator, show that where the chest is small and the supply pipes insufficient in carrying capacity, the pressure in the chest, tending to press the valve to its seat, is least during the period when the cylinder is taking steam; say, during about the first and fifth half quadrants of crank position of the average automatic cut-off engine, cutting off at one-fifth to one-fourth stroke when properly loaded.

Q. Does the pressure under a slide valve (tending to lift a slide valve from its seat, if it is free to move) vary during the piston stroke? If so, why?

A. The pressure under a slide valve, tending to lift it from its seat, being simply that exerted upon it through the end ports and exhaust ports, of course varies greatly during the stroke. Just what are these upward pressures per square inch, at various positions of piston, is easy to determine; but what are the valve areas thus exposed to upward pressure, is much more difficult to analyze, owing to the fact that the valve is shifting, and that its speed and direction, compared with those of the piston, are constantly changing.

INDEX TO PART II.

NOTICE.

In Part I. will be found a very copious Index referring to the four hundred questions answered therein; and that Index should be consulted at the same time as this one, no matter what the subject; as there are few topics relating to the steam engine which are not discussed in both volumes or parts.

[In looking for any subject having more than one word, look first for what appears to you to be the principal word, and if you do not find it under the initial letter of that word, try under that of the other.]

A.

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F.

H.

I.

PAGE

V.

W.

Z.

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